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THE

AMERICAN STEAM ENGINEER,

THEORETICAL AND PRACTICAL,

WITH

EXAMPLES OF THE LATEST AND MOST APPROVED
AMERICAN PRACTICE IN THE DESIGN AND
CONSTRUCTION OF STEAM ENGINES
AND BOILERS OF EVERY
DESCRIPTION.

For the use of Engineers, Machinists, Boiler Makers,
and Students.

BY

EMORY EDWARDS, M. E.,

AUTHOR OF "A CATECHISM OF THE MARINE STEAM ENGINE," "MODERN
AMERICAN MARINE ENGINES, BOILERS AND SCREW PROPELLERS,"
"THE PRACTICAL STEAM ENGINEER'S GUIDE," "MODERN
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THIS VOLUME
IS
DEDICATED
TO
J. FRANK MORRISON, ESQ.,
PRESIDENT OF
THE "CRESCENT CLUB,"
BALTIMORE, MD.

A noble, generous-hearted man,
Faithful to his friends,
Generous to his foes.
A Chivalrous Nature.
Sans peur et sans reproche.

AS A TESTIMONIAL
OF HIGH REGARD AND ESTEEM,
BY HIS FRIEND,
THE AUTHOR

PREFACE.

AT every hand, and at all times, whenever we take the trouble to exercise our powers of observation, evidence is presented of the absence of a knowledge of what might be termed "first principles" of mechanical cause and effect.

The extreme theorist, whose speculations are built almost entirely on the published opinions of those he calls "Authorities," does not take much trouble to assure himself as to the similarity of conditions existing in his own case and that of said authorities.

By way of side issue, perhaps it may be pertinent to remark that the variation of conditions under which any certain operation is performed, constitutes, in some cases, the slight mist that half obscures and magnifies, and in others the dense fog that leaves us totally uninformed as to our whereabouts.

Theory, which bears a somewhat similar relation to practice that steam does to water, is simply *the law of fact*, and must always be deduced from fact; if now we obtain a correct comprehension of the fact, and in a majority of cases we do not at the first obtain it, a course of reasoning may be followed, and a theory established, that will very materially aid us when we again observe a similar effect.

But when it is realized how often the reasoning faculties, unless subjected to a long and careful training, misconstrue the evidence which the senses receive and transmit, it is not a matter of wonder that so many wrong conclusions, based upon false premises, are continually being encountered.

At the risk of being charged with lecturing on "Intellectual Philosophy," the writer will remark that the senses of seeing, feeling, hearing, etc., are the instruments by which evidence is presented to the reasoning power, and if a wrong conclusion is drawn from the evidence, the fault lies with the reason, and not with the senses.

The ability of the reason to weigh each particle of evidence, and allot to each its correct value, is the gauge of precision by which minds must be compared.

This power of "exact valuation" can only be acquired by experience, however well cultivated the mind may be.

Amid the sound of moving machinery, the trained intellect will receive instant notice, by means of the sense of hearing, of any unusual noise, and judge of its whereabouts and cause with remarkable quickness and precision; but we will resume our subject.

At the other extreme of the line is the self-styled "practical man," who is so ignorant that he is unaware of his own short-comings, and who, after a certain age, becomes thoroughly case-hardened in his "views."

He, of course, has abilities in his own peculiar line, but arrives at most of his conclusions by the art of "guessing."

If he builds a machine he puts in sufficient metal to make two, making safe against break-down to be sure, but never entertaining the slightest suspicion of proportion; or he may make some parts so weak that it is necessary to "brace up" the contrivance with numerous sticks of timber, to keep it together when the belts are on.

Debating with the "practical man" is discouraging; his conclusions are all formed, and no possible discovery can modify them one whit; his mind is too inelastic to grasp new ideas; he meets reasoning with stubbornness, and however convincing an argument may be to a man who uses his wits, with the "practical man," whatever disagrees with his "creed," must be wrong somewhere. To sum up his weakness in few words, he is densely ignorant of *first principles.*

Between these two extremes of character is the man who "never grows too old or too wise to learn;" *he* sifts the information which is constantly coming to his senses, holds fast to that which is new or valuable, and promptly rejects that which is of no account.

He is, of course, his own judge of quality in the operation, but experience and observation are his trusty allies.

He studies the laws of nature, and settles in his mind as far as he can, the "first principles" of cause and effect with which his vocation brings him in contact, and when a difficult problem confronts him, he bases his efforts firmly on foundations that have been well tried and not found wanting; this mode of procedure is direct and to the point, and success follows as a matter of course. This book is intended for such men. A knowledge of the laws of ordinary natural forces and effects should form a part of the stock in trade of every mechanic; but a deplorable absence of such knowledge is constantly making itself manifest, and if these pages shall assist any seeker after "more light" to attain that end, the object of the writer will be fully accomplished.

In conclusion the writer begs to return his thanks to his friends, Wm. H. Hoffman; T. M. Goodeve; Chas. A. Hague; "Chordal;" and also to "The American Machinist," and "The Engineer" (of New York), for many favors.

Baltimore, Md., May 16, 1888.

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INTRODUCTION.

As this book may fall into the hands of some student who may not be "posted" in the First Principles of Mechanics, upon which so much depends for success in any branch of engineering, the following will be found a most useful and entertaining study, before he essays to grapple with what follows. It is intended especially for those who find, somewhat late in life, that they have neglected something. A vocation in life may be chosen, and in a few years one may find that he could be remarkably successful in that occupation, were it not for his ignorance of the laws which he finds himself compelled to yield to. Few, under such circumstances, can enter on a systematic course of study, and it is thought that the volume here presented may furnish some useful information, without the effort of dutiful research and study. Among apprentices in the shop the author has noticed a large percentage hungry for knowledge of the matters here treated. These boys, in many cases, are without common education, and their future success may depend on their ability to reason truth out of the vague fallacies of shop philosophy. The average text-book on physics seems to have been prepared especially as an accompaniment to other branches of study, and, in many cases, such text-books seem to have been prepared simply as a means for defining to an educated instructor a course of personal explanations to the students. The ambitious young man, thirsty for knowledge, gets little benefit from these books, which bear constant evidence of having been prepared for another class of persons.

It is hoped that the apprentice in the shop will find something interesting in these pages, and thus see that science is, after all, a something which he deals with every day, instead of the far-off dreamy nothing he is wont to think it.

GENERAL PROPERTIES OF MATTER.

MATTER is the term applied to everything which has size, substance, or weight. All matter is composed of very small particles held closely together. These small particles are called *molecules*, and the attractive force which holds them together is called *cohesion*. The molecules are composed of still smaller particles which are called *atoms*.

We can take almost any substance and break it up by some means and reduce the mass to fine powder. However fine this powder may be, we may still continue to break up each little particle into still smaller pieces. There seems to be no limit to the divisibility of matter, but science tells us that there is a stopping point somewhere; that there is a point in the division of matter where the particle is no longer composed of separate particles, and cannot therefore be further divided or separated. The term atom applies to these last indivisible particles of matter. Their size is inconceivably small, and their real separation may be considered a practical impossibility.

The division of matter is an every-day process in the arts. The mechanical process of dividing matter is illustrated in all kinds of powdering and grinding. A grain of wheat is a single piece of matter, but the act of grinding breaks the grain into two pieces, then breaks these pieces, and thus by successive stages divides each piece again and again until the single grain of wheat is reduced to a fine powder composed of countless thousands of pieces, which may even be further divided into still smaller pieces. Science tells us that each of these minute pieces is composed of countless atoms, and that when the division has reduced the mass to atoms no further division can take place.

We can now understand that there is no such thing as breaking matter. Instead of being broken, the separate particles are simply separated. What we say of the grain of wheat we may say of all masses of matter.

Another method of dividing masses of matter is by solution. If a piece of indigo, the size of a pin head, be put into a gallon of water, the indigo will break up into such very fine particles that each little drop of water in the gallon will be found to contain one or more of them, and each drop will receive a blue tinge. There are half a million drops of water in a gallon, and we thus see that our trifling piece of indigo has been divided into at least that many pieces. As fine as this division is, the particles are still visible to the naked eye, which accounts for the blue tinge of the water. If we pour this gallon of blue water into ninety gallons more, we have fifty million drops of water each containing one or more particles of the indigo. The blue tinge will then be too delicate to observe, but the subdivision of the indigo into smaller pieces can be carried very much farther still.

Evaporation is still another method of dividing matter. A minute quantity of musk, or other similar perfume, will separate into particles so small and numerous that every cubic inch of air in a large room will be perfumed by it. The air in the room may be replaced by fresh air continually, but this one little mass of musk will throw off particles enough to perfume this changing air for years. The number of particles into which the musk has thus been divided is past all conceiving.

All substances are composed of atoms, but these atoms do not touch each other. Around and among the atoms there lies a delicate, subtile fluid called *ether*. The space occupied by this ether is very much greater than the space occupied by the atoms. The spaces between atoms are called *pores*, and all substances are more or less porous. In some substances there are large visible pores, large enough to admit common air, which is itself, like all other matter, made up of atoms separated by ether. We can see no pores in close-grained iron or close-grained wood, or lead, or mercury, or in some hard clay ware, but in cork and sponge, and some kinds of wood and some very open kinds of iron, the pores are large and visible. If a cylinder be cast of great strength, of the closest iron, and a piston be fitted tightly to it, and water be put into the cylinder, and an immense pressure applied to the piston, water will find its way through the pores of the metal and will ooze in fine sweat from the outside. There are some substances which it seems hard to believe to be porous, such as glass, for instance, but science has determined that the pores form the bulk of all substances. In many substances the presence of pores and their great volume can be easily shown. A brick will hold a large amount of water and be no larger; the water simply enters the large pores. If a brick be powdered up very fine so that its particles can be packed very close together, it will be found that there is very little of it; and it will also be found that the water which a brick will hold has a great deal more bulk than this closely packed brick. A sponge is another illustration. Liquids have pores like all other substances. Liquids are made up of little atoms and larger pores, and the atoms and pores in different liquids are of different sizes. In some liquids the pores are large enough to admit the atoms of other liquids. In this case it will be seen that a liquid may actually hold more liquid without increasing in bulk. It would be in some degree like pouring water into sand. A gallon of water and a gallon of sand mixed will not make anything like two gallons of the mixture.

In the same way, strange as it might seem before the matter was somewhat understood, a gallon of water and a gallon of alcohol mixed will not make two gallons of mixture.

The small particles, of which all matter is composed, are drawn toward each other by some inner natural force, which we call *cohesion.* In some substances this cohesion is so small that we may easily pull the particles apart. In some others the cohesion is so great that enormous power is required to overcome it.

The particles composing a piece of clay may be pulled apart very easily; but two strong men pulling on an iron wire, a sixteenth of an inch in diameter, cannot overcome the cohesion of the particles and break the wire. It will thus be seen that the strength of any material depends on the cohesive attraction between the particles. The cohesion and consequent strength of some materials may be altered by working in various ways. An ordinary piece of bar iron may have a certain strength, but the operation of working the iron and drawing it into wire will sometimes increase its strength five-fold. The various operations probably crowd the particles closer together, and thus allow the cohesive force to act at shorter distances.

Besides the force of cohesion, there is an opposite force called *repulsion*, which, when in action, tends to throw the particles of matter away from each other. In fact, the two forces are always acting against each other in all substances. Were it not for repulsion, the atoms would rush together and touch each other, and all matter would become solid. These two forces, always acting, are constantly changing their relative powers. The application of heat to substances increases the repulsion, the particles move further apart, and the mass of matter becomes larger. This is the common action we speak of as expansion by heat. It will be seen that in reality the mass itself does not expand; the particles forming the mass are simply moved further apart. The force of repulsion, like the force of cohesion, is enormous. In most substances, repulsion develops but slowly, while in others it is instantaneous. Thus certain substances, properly treated and mixed, form gunpowder, and the application of high heat to gunpowder will develop repulsion between the particles. They fly apart with immense force, and this force we often make use of in peace and war.

A change in the internal forces of matter often changes the nature of the substance. A piece of ice one inch square, for instance, has great cohesion. It is hard to pull the particles asunder. The application of heat to the ice will separate the particles so far that the force of cohesion will no longer hold them in place. They will roll around and fall and slide over each other. By this application of heat, the substance has been changed from a solid to a liquid. There are, however, still the same number of minute particles, and their form and nature has not been changed. If we apply heat still further, we separate the particles still more, and finally they become pushed apart with a force which will lift fifteen pounds for each square inch which these particles act upon. This takes place when the thermometer indicates 212 degrees. It is the boiling point of the water, which now changes from water into a vapor, and the cubic inch of ice has been expanded into seventeen hundred cubic inches of vapor. If this vapor was confined in a vessel, instead of being allowed to expand freely, the repulsive force would be so great as to burst the vessel, if not of great strength. If we take the heat away from this vapor, the particles draw together again. This action is termed *condensation*, and when the heat has been reduced to below 212 degrees, we again have water, or the matter in a liquid state, and when the heat is reduced to below 32 degrees, we have a cubic inch of ice again, and our substance is once more in the solid form. No change has been wrought upon the particles of matter ; they have simply, by changes in the forces of cohesion and repulsion, been separated from each other, and again brought near to each other.

Just such changes take place in many substances by the application of heat. Iron, at ordinary temperatures, is a solid; at a high heat it softens; at higher heat it turns liquid, and at still higher heat it vaporizes; and by the removal of heat it is brought back successively to liquid and solid.

In some substances the particles are held together with such a light cohesive force that they may be pushed apart with ease. When we can do this, we say the substance is *soft*, as putty or clay for instance; but when the particles insist on holding their usual position, and yield only to great force, we say the substance is *hard*.

In stone and the metals, for instance, we move the particles with great difficulty.

A finger may be punched into putty with ease, but powerful machinery is required to disarrange the particles of matter forming iron.

Substances which crumble and break, instead of changing form, when force is applied to them, are called *brittle*. Chalk, stone, glass and cast iron are examples; but substances which permit a change in form are called *malleable*. Gold, wrought iron, lead, platinum and copper, are examples. In malleable substances, the small particles can be moved without separating so far as to be beyond the range of cohesive force. Some brittle substances become malleable when heated. Wax is such a substance, and glass, when heated, can be spun into the finest and most delicate threads.

Substances which can be drawn into wire are called *ductile* substances. Many of the metals are ductile to a great degree, while some are not at all so.

Gold is very ductile and very malleable. It can be drawn into the finest wire and can be hammered into sheets so thin and delicate that it takes two hundred and fifty thousand sheets to make a pile an inch high.

Almost all substances possess some peculiarity in structure; some variation in the size of molecules; some variation in the distance between them; and, consequently, some variation in their cohesion or strength. Many substances may have their structure altered by various processes, and thus take a new nature.

If we examine a piece of soft steel, we notice that it is composed of rather coarse grains. If we strike this steel with some hard substance, we find that the particles are moved, and the steel is bruised and dented. By means of a file we can even pull particles away from the mass and reduce its size. We judge by this that the cohesion of soft steel is not extraordinarily high. The particles do not strongly object to being moved out of their usual position. We may bend the piece of steel, and thus change the relative position of every particle, and the piece will stay bent, thus indicating that the particles are not tightly bound together. If we take this same piece of soft steel and heat it till red, and then suddenly cool it in water, or other cold matter, we will find that the molecular structure is greatly altered. If the piece be broken, it will be noticed that the grains are very fine and very closely compacted together. The cohesion of the particles has increased so very much that it is found almost impossible to separate them. Hard blows with the hardest of substances will not move the particles so as to form dents.

The hardest of files will not separate any of the particles from the mass. It is now hard, and its cohesion is so great that it will move the particles of almost all other substances against which it may be forcibly applied. This quality fits hard steel for tools, etc.

If an attempt be made to bend this hard steel, it will be found that the particles are bound together so tightly and resist a change of position so powerfully, that they will not move until sufficient power is applied to separate them entirely. The piece of steel breaks instead of bending. Hardness is always accompanied by brittleness. If we heat this hard steel and cool it very slowly instead of suddenly, we again have soft steel instead of hard steel; malleable steel instead of brittle steel; particles far apart and loosely held, instead of closely packed and powerfully held together by cohesion.

Copper is exactly the reverse of steel in the way its cohesion is affected by sudden heating and cooling. Soft copper heated red and suddenly cooled becomes still softer and more ductile and malleable. Soft copper heated and slowly cooled becomes harder and more brittle.

Glass is a very hard and brittle substance. Its particles can be moved on each other only by a harder substance. Diamond is very much harder, being, in fact, the hardest substance known.

Hard substances, owing to their brittleness, can often have their fine molecules broken away by light blows. A highly polished piece of the hardest steel, if exposed to the action of a dusty wind, will become dull and rough by reason of the outside particles being hammered and cracked off by the millions of light blows received from the flying particles of dust. The diamond is so very hard, and its particles so extremely cohesive, that it withstands for many years the hammering action of the dusty air. A false diamond, made of the hardest of glass, and polished in the same way and as highly as a real diamond, will reflect the light with great brilliancy and will appear like a diamond, but a few hours' exposure to the common air serves to break away and deaden its comparatively soft surface.

As before stated, the combination of two different substances often produces a substance unlike either. Thus copper and tin are both very soft, ductile, and malleable metals. Nine parts of copper and one part of tin, melted and mixed, form the alloy known as gun metal. This alloy, unlike either of the metals of which it is formed, is very hard and tough, and possesses comparatively no malleability or ductility. Of the two metals composing this alloy, tin is much the softer, yet an increase in the proportion of tin in the alloy *hardens* it. When the alloy is one-third tin and two-thirds copper, it is so hard that steel tools will not cut it, and so brittle as to fly to pieces when subjected to light blows.

In speaking of the molecular change in steel when suddenly cooled after being heated, it was stated that soft steel allowed its molecules to be moved around on each other by denting or bending, and that the molecules will stay in any new position where put. Thus a blow on soft steel leaves a dent, or a new arrangement of the molecules. When the steel is hard, no dent can be made, because the molecules refuse to move.

If, instead of cooling the steel suddenly, it be cooled neither suddenly nor slowly (a process called tempering) we will have a new condition of the molecules, and new properties in the steel. It will be neither very hard nor very soft. If it be struck with a hard substance, the molecules will move, but will immediately return to their old positions. The steel can be dented, but the dent instantly disappears. When the steel is bent it instantly straightens again with considerable force, Steel in such condition is fitted for springs. When a material is composed of molecules which will allow themselves to be displaced, and still return to their original positions, that material is said to be *elastic.* Glass, though hard and brittle, is very elastic. Ivory is very much so; india rubber is elastic to a remarkable degree, and spring steel, and spring tempered metals generally, are very elastic. If a lead ball be dropped on the pavement, it will be flattened on the side which strikes; the flat part will stay flat, because the lead has no elasticity. If an india-rubber ball be dropped on the pavement, it will be flattened very much more than the lead ball; but the elasticity of the substance is so great that the particles all resume their old positions with such force as to bounce the ball upward to a considerable height, and when the ball is examined, no sign of the flattening is seen.

Some substances may be squeezed or compressed into very much smaller bulk; such substances are said to be compressible. If, upon being released, they return to their former bulk, they are of course elastic. All substances are to some extent compressible. Under great pressure, the ether is crowded out from between the atoms, and the substance becomes more solid or *dense.* The real *matter* in substances is so little, compared with the ether between the atoms, that it has been stated that the whole earth, if the atoms, were pressed into actual contact, and made perfectly dense, would not fill a square foot of space. Such a statement seems hard to believe, but will serve to bring our minds to a sense of the *porousness* of all matter.

A cubic inch of lead can be compressed into one-tenth of that bulk, and it will remain so, and be more dense or solid. A room-full of air can be compressed into a thimble, but will immediately expand to its full volume when released, owing to its great elasticity.

We know very little of the real cause of natural forces. We know that some great force tends to draw the atoms of some matter closely together, and we call that force *cohesion.* We know that some great force tends to force the atoms of some matter apart, and we call that force *repulsion.* While we know little of the cause of these forces, we may, by close observation, become familiar with their general laws of action, and it is a knowledge of these general laws of nature which forms the basis of science and of all practical operations with matter.

In many substances, as has been stated, some natural force causes one atom to draw closely and powerfully toward other atoms.

There is also another natural force which tends to draw all atoms and all substances, in fact everything, toward the centre of the earth. This force we call *gravity.* Cohesion draws the atoms composing a piece of iron closely together with great force, and gravitation draws the piece of iron powerfully toward the centre of the earth. It seems as if all pieces of iron, and all other pieces of matter, were simple atoms constituting a great universe which the force of gravitation was seeking to bring to the centre of the earth. If we lift a weight it falls, and if it could fall far enough, its line of travel would pass through the earth's centre. A plumb line suspended gives us the direction to the earth's centre. Two plumb lines five miles apart, would each point to the earth's centre, and as the earth is round, these two lines would not be parallel. They would be further apart at the top than at the bottom, and would cross at the centre of the earth. A very long building, whose walls are perfectly plumb, is wider at the top than at the bottom. A little study will show that this tapering building with the large end upward is the most secure form. The stones composing it tend to fall to the earth's centre, and as the walls are built in that direction, it follows that each stone is supported by other stones placed directly under and in the real path of the force of gravity.

Gravity acts with a different degree of force on different substances. Those substances which are attracted powerfully we call *heavy* substances, and those which are attracted but slightly we call *light* substances. In comparing heavy and light bodies we speak of their *weight.*

In referring to the size of bodies, science uses the term *mass*, which means the same as bulk. A large thing is spoken of as a thing of great mass.

In studying the action of gravity we must consider it as acting on substances of equal mass. Thus it would not be proper in investigating the laws of gravity to experiment on one cubic foot of lead and two cubic feet of iron. The mass of the iron would then be twice the mass of the lead, and we could determine but little as to the laws of gravity.

We should deal with equal masses and study closely the different effects of gravity on similar masses of different substances. As before stated, when the atoms of a substance are packed closely together we say that substance is *dense.* If a substance is very dense, we can readily see that there is more matter than ether in it; while if it is very porous there may be very little matter in a large mass. Gravity acts on substances in proportion to their densities.

Lead is a very dense material; gravity acts on it powerfully and we say it is heavy. Cork is very porous; gravity acts on it slightly and we say it is light.

A cubic foot of cork weighs fifteen pounds, and a cubic foot of lead weighs 700 pounds. We say that lead is more than forty-six times as heavy as cork. There is so much difference in the density of these two substances that we can easily see it. We see great pores in the cork, and thus know that the particles composing it are not packed closely together as they are in lead. Still the mass of the lead and the cork is the same, for we are speaking of a cubic foot of each. There is the same difference in nature between all substances as we see in these two substances, but the degree of difference is not the same. Thus cast iron is very much denser and heavier than cork, but is lighter and more porous than lead.

If we should compress a cubic foot of cork with sufficient force we might make it very dense, and reduce its mass so that forty-six cubic feet would occupy only one cubic foot of space. If we should do this the cork would become as dense as lead, and its weight would be the same.

Gravity acts on all matter. There is no substance which does not have weight. If we take an evenly-balanced pair of scales and place a cubic foot of lead in one pan and a cubic foot of cork in the other pan, the lead will be drawn downward by gravity; the cork will also be drawn downward by gravity, but the superior attraction for the dense lead so far overbalances the attraction for the porous cork that the cork rises and the lead falls. Iron dropped into water sinks and crowds water upward, while iron dropped in quicksilver is held upward and floats on the heavier quicksilver.

Almost all substances will drop through the light air, but common gas is lighter than air, and floats upon it. The air is the heaviest, and is attracted the most by gravity. All these substances, however light and airy, have some weight. We look upon the air as a something having no weight, because we are always comparing it with substances very much heavier. Air has weight, and a bottle full of it will weigh more than a bottle from which all air has been exhausted. Air weighs seven one-hundredths of a pound to the cubic foot. It would take 10,000 cubic feet of air to weigh as much as one cubic foot of lead.

We have before noticed that the closer together the atoms of matter are packed, the greater the cohesion. This is also true of gravity acting on matter. The closer matter is brought to the centre of the earth, the more it is attracted by gravity. Gravity acts with much greater force at the bottom of a deep mine than at the top of a high mountain.

We have thus far considered only a few of the properties of matter, but they have been very important properties, a knowledge of which is necessary to the intelligent exercise of ordinary every-day operation with materials. Other properties of matter will be spoken of further along in these pages. We have looked into the structure of matter, and find that the peculiar arrangement of molecules in different substances gives to those substances different qualities of *weight*, *hardness* and *strength.*

The properties of almost all substances have been determined by practical scientific investigation, and it is thought well to insert at this point tables showing the different relative properties of different well-known substances.

WEIGHT OF DIFFERENT SUBSTANCES.

The following table shows the weight in pounds of a cubic foot of various common substances. From this table we can calculate the weight of any mass of any substance in the table, by first ascertaining by measurement and calculation how many cubic feet there are in the mass, and multiplying the result by the weight of a cubic foot of the substance:

SUBSTANCE.	Pounds per cubic foot.
METALS.	
Cast iron	450.5
Wrought iron bar	486.6
Soft steel	489.8
Common brass	537.7
Wrought copper	555.
Cast lead	709.
Cast zinc	428.8
Tin	456.
Mercury, at 60°	848.75
Gold	1013.
Silver	551.
Antimony	422.
Aluminium	160.
Platinum	1296.
WOODS.	
Oak	55.
Ash	53.
Cork	15.
Cedar	35.
Chestnut	38.
Hickory	45.
Lignum-vitæ	83.
Mahogany	50

Pine, yellow	34.
Spruce	31.
Walnut	31.
Willow	30.
MISCELLANEOUS.	
Air	00.075
Brick	102.
Coal, soft	80.
Coke	62.
Pressed cotton	22.
Gutta-percha	61.
Ice, at 32°	57.5
Moist sand	128.
Common earth	137.
Gravel	126.
Glass, window	165.
Granite	165.
Grindstone	133.
Marble	169.
Millstone	155.
Mud	101.
LIQUIDS.	
Alcohol	50.
Milk	64.
Petroleum oil	55.
Water, rain	62.5
" Dead Sea	77.5

STRENGTH OF SUBSTANCES.

The following table shows the breaking strength of, or power in pounds required to pull apart a bar one inch square of various common substances. The strength of any piece of substance shown in the table can be found by measuring or calculating the number of square inches area of the piece and multiplying the result by the breaking strength of one square inch, as shown in the table:

SUBSTANCE.	POUNDS.
Wrought copper	34,000
Copper wire	61,200
Cast iron—American	31,800
" English	19,500
Iron wire	103,000
English bar	56,000
Boiler plate	50,000
Cast lead	1,800
Platinum wire	53,000
Cast silver	40,000
Bar cast steel	88,000
Shear steel	124,000
Cast tin	5,000
Zinc, cast	3,500
Hydraulic cement	234
Glass	2,350
Gutta-percha	3,500
Ivory	16,000
Leather belts	330
White marble	9,000
Mortar, good	60
Manila rope	9,000
Hemp rope	15,000
Wire rope	37,000
Stone	350
Gold	40,000
Brass	42,000
Bronze, best	56,800
Oak	10,000
Boxwood	20,000
Beech	19
Maple	14

Yellow pine	14
Cedar	13
White pine	11
Hemlock	8
Cork	5

HARDNESS OF SUBSTANCES.

It is, of course, impossible to speak of hardness in pounds, or square feet, or anything of that kind. We can only compare the hardness of one substance with the hardness of another substance. Thus, we can say that chalk is half as hard as gold. It is best to take some one substance as a standard and compare other substances with it. Taking Ormuz diamond as the standard, we find cork to be one-twentieth as hard.. For convenience it is best to give the standard a value of one hundred, and express the hardness of other substances in hundredths of the standard. The following table shows the relative hardness of various substances, Ormuz-diamond being taken as the standard:

SUBSTANCE.	HARDNESS.
Ormuz diamond	100
Agate	71
Quartz	70
Ruby	64
Granite	22
Chalk	15
Hard steel	65
Unhardened steel	40
Cast iron	38

Wrought iron	37
Platinum	47
Common brass	32
Hammered gold	30
Hammered copper	34
Hammered silver	32
Zinc	26
Aluminium	24
Tin	24
Babbitt metal	20
Cast lead	15
Ebony	24
Boxwood	22
Oak, hard	22

CHAPTER I.

THE THEORY OF THE STEAM ENGINE

THE theory of the steam engine is too extensive a subject to be treated here, except in a very concise manner. We only attempt a plain and practical statement of the course pointed out by science to be pursued in our desire to measure the economical efficiency of the steam engine.

To those who desire to go deeper into the matter, we would recommend "Text-Book of the Steam Engine"—a most masterly work by T. M. Goodeve, of London, England, a compact and clearly put treatise.

The following will be sufficient for all practical purposes. (Theory and Practice, like Capital and Labor, are twin sisters which should ever go hand in hand to achieve the best results. Neither can do without the other in keeping step to the world's wonderful progress.)

In 1824 Sadi-Carnot, who was a French officer and a son of Carnot the great war minister of Napoleon I., demonstrated that the efficiency—that is to say, the power of doing work—of any steam, gas or hot-air engine, depended solely on the difference between the temperature T, at which it received the fluid, and the temperature t_1 at which it was rejected, and this principle which he then enunciated is a fundamental truth according to the dynamical theory of heat. It is a very common thing to hear some one get off something like the following:

"One pound of coal develops in its combustion 12,000° Fah. units of heat which in their turn represent 12,000×772 foot pounds of work or 9,264,000 foot pounds, and these represent a consumption barely ¼ pound of coal per indicated H. P. per hour, whereas few engines produce an indicated H. P. per hour with less than ten times that expenditure, or say 2½ pounds of coal."

Just so. Now a pound of coal can only do work by the operation of a heat engine (and a steam engine *is* a heat engine, for it is the *heat* that does the work and nothing else), its function being to store up heat in a gas (the steam), which does work by expanding between *two given temperatures.*

Let a pound of coal supply without any loss steam at say *300* pounds pressure to a *perfect* engine where the condenser is at a temperature of 100° Fah. The temperature of the steam will be 417° Fah. Hence work done $=\frac{877-560}{877}=\frac{317}{877}=\frac{4}{11}$ I. H. Now I. H. represents $1\frac{7}{10}$ H. P. for *one* pound of coal. 2½ pounds of coal for H. P. is the same as one pound of coal for $\frac{2}{5}$ H. P. Hence the comparison lies between one pound of coal for $1\frac{7}{10}$ H. P. and one pound of coal for $\frac{2}{5}$ H. P., which is a little more than 4 to 1 instead of 10 to 1. We begin by throwing away $\frac{7}{11}$ of the whole heat, which goes into the condenser and is lost; we are compelled to work between tempertures which lie between moderate limits of difference, and hence it is useless to refer to the whole heat as a standard of comparison. To get all of the work (heat) it would be necessary to expand down to the temperature of absolute zero, which is about —460° Fah. This is practically and utterly impossible, and that condition is as hopelessly removed from our reach as if it were actually in some distant planet.

The progress made in the economy of fuel by successive improvements in the steam engine and boiler can be readily traced by looking back historically and comparing the old with the new.

In 1775 it was about 27 pounds of coal per H. P. per hour, Smeaton brought it down to 17 pounds of coal per H. P., and Watt to 8 pounds coal per H. P., and this (8 pounds) is about the average of the present "wild-cat" stationary engine and boiler in general use all over the country in this year 1887. It is needless to say that none of *that* kind are represented in this book.

In 1863 the compound engine brought it down to about 4 pounds of coal per H. P. per hour. Some of our best engine-builders have brought it down as low as 2 pounds per H. P. per hour. The greatest advance ever made, as yet, has just now (1887) been accomplished by Mr. James Howden, of Glasgow, Scotland. He has just fitted a steamship (the "Ohio") with triple expansion engines which develop one I. H. P. per hour on 1¼ pounds of coal. For this great feat Mr. Howden well deserves to be knighted by her Majesty the Queen, and so say we—all of us American engineers.

The man who will bring it down to one pound of coal per I. H. P. per hour will confer a boon on mankind. This done the engineers will have done all that can be hoped for or reasonably expected of them.

THE DYNAMICAL THEORY OF HEAT.

The dynamical theory of heat, which assumes that heat is a mode of molecular motion, affords a satisfactory explanation of various phenomena. In all cases of friction, compression and percussion, a certain amount of mechanical force is arrested, the energy of its visible motion is spent in producing molecular motion, and is thus transformed into heat.

The quantity of heat evolved is in proportion to the mechanical force expended.

Thus, when air is compressed, the rise in temperature is due to the mechanical effect or work which must be spent in driving the particles of the air nearer together. Conversely, *Heat is consumed in effecting mechanical work.* Let a cylinder filled with compressed air be cooled to the temperature of the surrounding bodies; its elastic force is competent to produce mechanical work, (1) by moving a piston, or (2) in displacing the air in front of the cylinder.

If now this air is allowed to expand into the atmosphere, the air will be chilled, because mechanical work has been performed by the expenditure of the heat to which the elastic force of the air was due.

The relation which exists between heat and work is known as the *mechanical equivalent of heat*, or simply as Joule's equivalent. To determine for gases, suppose a tall cylindrical vessel (Fig. 1.) C., whose section is equal to a square foot, and let P P be a piston without weight moving in the cylinder. If the piston be placed so that the height A P is one foot, it will inclose a cubic foot of air.

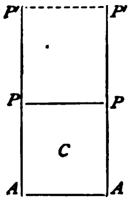


Fig. 1.

Now, if this air be heated 490° F., its volume will be doubled, and will raise the piston one foot, or to P′ P′. In raising it has overcome the pressure of the atmosphere above the piston, or has lifted 15 × 144 = 2160 pounds one foot, and has performed work equal to 2160 foot pounds.

With the same amount of heat, only about one-fourth as much water would have been raised 490°, because the specific heat of air is 0.24 that of water.

The weight of a cubic foot of air is .08 pound, hence the heat imparted to perform the work of the air, would have heated only $0.08 \times 0.24 = .0192$ pounds of water 490°.

This is equivalent to 9.4 pounds of water heated 1° Fah. Hence 9.4 thermal units have been required to raise 2160 pounds one foot high by the exhaustion of the air.

If the piston had been fixed so as to retain the air at a *constant volume* while being heated, the quantity of heat required to raise its temperature 490° would have been less than when expanding under a *constant pressure* in the ratio of 1.421 : 1.

Hence the thermal units required to raise the temperature of the cubic foot of air where kept at a constant volume is found to be $9.4 \div 1.421 = 6.6$ units. Deducting 6.6 units from 9.4 units, we find that the excess of heat imparted to the air when permitted to expand, is competent to raise 2.8 pounds of water 1° Fah.

This excess has been employed in performing the work of lifting 2,160 pounds one foot high. Dividing 2,160 by 2.8, we find that the quantity of heat required to raise one pound of water 1° Fah. is competent to lift 772 pounds a foot high. *This is, therefore, the mechanical equivalent of one thermal unit.*

In calculating the mechanical equivalent of heat, all the possible factors must be found and allowed for in order to obtain the exact equivalent. When a body falls freely through the air, a portion of its force will be expended by friction of the air, and the heat produced will be dissipated by radiation. If a body falling through a vacuum is suddenly arrested by collision with another, the heat generated will be in proportion to the height of the fall.

A portion of this heat may be again instantly converted into the mechanical motion of the rebound, and the remainder will be divided between the two bodies.

Now, since the height through which a body falls is proportioned to the square of the velocity attained ($V = 8.02 \sqrt{H}$), the heat generated by the percussion of bodies moving in any direction will be as the *squares of their velocities* at the time of impact.

The dynamical theory also explains the evolution and consumption of heat which accompany changes in the volume or state of bodies.

Thus when heat enters a body, its actual energy is absorbed (1) in increasing the intensity of molecular motion, which is shown by a rise in the temperature, (2) in separating the molecules, as shown by expansion, and (3) in re-arranging its molecules, or causing a change of condition. The work performed is partly internal and partly external. The exterior work is employed in overcoming external forces which resist the expansion, and the interior work is employed in separating and re-arranging the molecules within the mass of the body, by overcoming cohesion or affinity.

In whatever way we view it, the interior work performed by heat is enormous.

Thus, in expansion, a slight rise in temperature will produce a dilatation which would require the expenditure of tremendous mechanical power. Latent heat is merely a *consumption of heat* proportionate to the interior work required to *overcome cohesion* in melting or vaporizing a body.

The heat required to melt a pound of ice is one hundred and forty-three thermal units, which is equivalent to 110,396 foot-pounds.

The heat required to change boiling water into steam is 967 units, which is equivalent to 746,524 foot-pounds.

The actual energy of the heat may thus be measured by its equivalent of mechanical motion. Inasmuch as this motion may be again transformed into sensible heat by a *contrary change of state*, the atoms are said to possess a possible, or *potential energy.*

Thus, when a gas is liquefied by compression, external work is supplied, the interior work due to the cohesive force which draws the molecules together is transformed into sensible heat.

In like manner, the cohesive force which changes a liquid to a solid, performs interior work, and the potential energy becomes actual: that is, the latent heat becomes sensible. So also when two bodies unite by chemical affinity, the molecular motion is transformed to heat.

Thus, when $\frac{1}{9}$ of a pound of hydrogen combines with $\frac{8}{9}$ of a pound of oxygen, one pound of steam is produced, and 6,892 thermal units are evolved, which are equivalent to 5,320,624 foot-pounds.

The molecular force evolved in changing a mixture of these gases to a pound of ice will therefore be:

	THERMAL UNITS.	FOOT POUNDS.
1. Potential energy of combination .	6,892	5,320,624
2. " " steam	967	746,524
3. " " water	143	110,396
Total energy	8,002	6,177,544

This is equivalent to the force required to raise one ton to a height of 3,098 feet, or 3,098 tons one foot high.

Molecular forces are, therefore, by far the most powerful of any with which we are acquainted.

CHAPTER II.

ON THE BURNING OF FUEL IN THE FURNACE OF A BOILER.

IT being agreed that heat is the agent which does work in an engine, and that steam, air, and vapor are but the instruments for transmitting the motion of heat to the machinery, our object will be to store up in an elastic working substance the heat derived from fuel, and to guard against loss as far as possible.

As a general rule *chemical combination* is accompanied by the evolution or production of heat, and chemical decomposition by the disappearance of heat equal in amount to that produced during the previous combination of the elements which are undergoing separation.

Combustion, or burning, is the name given to rapid chemical combination attended with the evolution of intense heat.

It is necessary to bear these facts in mind in estimating the heating effect of fuel. Thus, where hydrogen and oxygen exist in coal in the proportion necessary for forming water (viz: one of hydrogen to eight of oxygen by weight,) it is usual to assume that they do not influence the heat of combustion.

The hydrogen is taken to have been already burnt in oxygen. In coal there may be 5 per cent. of hydrogen, and 4 per cent. of oxygen: this would leave 4½ per cent. of hydrogen available for heating purposes. There appear to be exceptions to the above rule, and Dr. Percy gives the result of an experiment where two coals closely agreeing in ultimate composition have been found to differ by 5 per cent. in heating power.

The composition of various kinds of coal is given by Dr. Percy, in his work on fuel, and it is well known that the differences in the constituent parts of coal are very great, and give rise to qualities of various kinds which influence the selection to be made for heating purposes.

The heat given out in the burning of hydrogen and carbon is estimated as follows:

	UNITS OF HEAT.
1 pound of hydrogen consumes about 36 pounds of air and gives out	62,032.
1 pound of carbon burnt to carbonic oxide, about 6 pounds of air and gives out.	4,452.
1 pound of carbon burnt to carbonic acid, about 12 pounds of air and gives out . . .	14,545.

According to Dr. Percy the *heating power* of a substance is the number of units of heat produced by the combustion of a unit of weight of the substance; and if the unit of heat be defined according to the Centigrade scale—

Heating power of hydrogen is 34,462.

Heating power of carbon is 2,473, when burnt to carbonic oxide.

Heating power of carbon is 8,080, when burnt to carbonic acid.

Also 1 pound of hydrogen evaporates 64.2 pounds of water at 212° Fah.

Also 1 pound of carbon (burnt to carbonic oxide) evaporates 4.6 pounds of water at 212° Fah.

Also 1 pound of carbon (burnt to carbonic acid) evaporates 15 pounds of water at 212° Fah.

It does not appear that the absolute heat of combustion can be increased, but it is easy to pile up the waves of heat in an enclosed space, and thereby to increase wonderfully the apparent power of the combustion.

The furnace may be looked upon as a large chemical apparatus in which coal and air are to be mixed together in the proportion best adapted for burning the fuel without waste. In performing this operation, an engineer falls very far behind a scientific chemist when operating on a small scale in his laboratory.

Thus a chemist in burning one pound of ordinary coal in a carefully protected chamber, would cause the heat from the fuel to evaporate (say) 14 pounds of water, whereas the evaporation per pound of coal in a steam boiler seldom exceeds 10 pounds, or 10½ pounds, of water, a common performance being the evaporation of from 6 pounds to 8 pounds of water.

Looking at the question as one of admixture of fuel and air, the rough numbers usually quoted on the authority of Rankine are the following: For the actual burning of ordinary coal in a furnace, 12 pounds of air are required in order to combine with the constituents of each 1 pound of coal.

But the gaseous products of combustion must be largely diluted, otherwise the air would not get at the fuel, and for this dilution as much air again is required, making a supply of 24 pounds of air to each 1 pound of fuel.

Thirteen cubic feet of air, at 60° Fah. and under a pressure of 30 inches of mercury, weigh about 1 pound. Therefore 312 cubic feet of air are required for each 1 pound of fuel, which comes to nearly 700,000 cubic feet of air for the effective burning of one ton of coal.

That gas and hydrocarbon vapor proceeding from coal require a good supply of air for burning was frequently shown by Faraday in a lecture experiment, and his illustration goes to the substance of the whole matter.

The device was to soak a little cotton-wool in any hydrocarbon liquid, and set it on fire in a jar of oxygen gas. In such a case the hydrogen devours the oxygen and the flames light up with dazzling brilliancy, but very soon the supply of oxygen fails, the light becomes less, when all at once, for no apparent reason, the burning wool throws out a dense mass of black flakes, which fill the jar in a thick cloud.

The quantity of soot ejected would surprise any one but a chemist, as few would be aware that the unburnt liquid was capable of throwing out such a supply of carbon.

It is needless to say that the effect here produced in the jar of oxygen is the same as that occurring in the chimney of a steam boiler when the supply of air is defective, the result being that so frequently seen, viz.: the pouring out of dense black smoke into the atmosphere. The loss of heat from unburnt gases may also take place without being made evident by the issuing of smoke. Thus carbonic oxide may pass away instead of carbonic acid.

There have been a great number of inventions relating to the prevention of smoke in steam boilers which cannot be discussed in the space here available.

Smoke once formed cannot be burned. The proper thing to do is to devise means to prevent the formation of smoke.

The various modes in which fuel is wasted have been classified by Rankine somewhat as follows:

1. Fuel is lost by the escape of gases in an unburnt state, or by permitting black smoke to be thrown off.

Here the supply of air is defective, and the physical action is traced to the remarkable affinity of hydrogen for oxygen gas, whereby the oxygen is absorbed to the exclusion of carbon in the first instance.

2. There is waste from external radiation and conduction. M. Peclet states that the quantity of heat radiated from incandescent charcoal is 5% of the total heat of combustion, and that the heat radiated from coal somewhat exceeds that radiated from charcoal. The practical conclusion to be drawn from this statement is, that the heat radiated from the burning fuel should be carefully intercepted in every direction.

Hence the economy resulting from the use of an internally fired boiler with internal furnace tubes.

As to the heat radiated into the ash-pit, that is carried back again to the fire by the current of entering air. In respect of the loss of heat by conduction, that is obviated as much as possible by the use of fire-brick, and where the furnace is outside the boiler, the resistance to conduction is increased by double layers of brickwork with enclosed air spaces between the layers.

3. There is loss of heat by the escape of gases up the chimney at a temperature above that which is necessary for maintaining the draught.

A general idea of the value of a chimney in promoting the draught of a fire may be gathered from a statement of a law which appears to be approximately true, viz: That the velocity of air, as due to increased pressure, is that acquired in falling down a height equal to the uniform column which gives the increased pressure.

In making any calculation on this subject it is usual to adopt the hypothesis that air is incompressible and behaves as a liquid.

Let the increase of pressure support 5 inches of water. We know that 29.922×13.596 inches of water balance the pressure of the atmosphere which would be produced by a stratum of incompressible air 26,214 feet high.

Therefore, 1 inch of water will balance 64.4 feet of air.

Hence 5 inches of water balances 322 feet of air: therefore velocity due to increase of pressure

$$=\sqrt{64.4\times322}$$
$$=64.4\sqrt{5}$$
$$=144 \text{ feet per second, very nearly.}$$

According to the old rule the area of the chimney should be $\frac{1}{10}$ that of the fire-grate, and there should be 1 square foot of fire-grate for each horse-power.

The consumption of coals per horse-power being estimated, Mr. Bourne gives Boulton and Watt's rule for proportioning the dimensions of land chimneys, according to which a factory chimney 80 feet high would have a sectional area of 400 square inches, the consumption of coal being 300 pounds per hour, and the suction of the chimney being that due to a pressure of a little more than 1 inch of water.

Rankine gives formula for computing the height of a chimney in order to produce a given draught, and states that the best chimney draught takes place when the absolute temperature of the gas in the chimney is to that of the external air as 25 to 12, or when the density of the hot gas is one-half that of the external air.

For example, if the temperature of the external air be 50° Fah. the best temperature of the hot gas in the chimney will, according to this rule, be 602^c Fah., which is less than that of melting lead, viz., 620° Fah.

Hence the rule that to insure the best possible draught in a chimney the temperature of the hot gas should be nearly sufficient to melt lead.

If the temperature of the furnace itself be estimated at 2,400° Fah. and that of the issuing gases at from 600° Fah. to 700° Fah. or even higher, as is often the case, we see that 25 per cent. of the heat of combustion passes up the chimney and is consumed in producing a draught of air through the furnace grate.

The loss of heat from the waste gases may be lessened by the use of an economizer for heating the feed water.

Sir W. Fairbairn, in his treatise on "Mills and Millwork," describes an economizer introduced by Mr. Green, of Wakefield, as consisting of a series of upright tubes forming a supplementary boiler placed in the main flues, and states that the formation of soot on the pipes was the source of the ill success of previous attempts in this direction.

This difficulty has been overcome by an apparatus of scrapers or cleaners, and it is found that when the waste gases escape at a temperature of 400° Fah. or 500° Fah. the feed-water can be heated to an average of 225° Fah., the temperature of the gases after leaving the pipes being reduced to 250° Fah.

To produce this effect 10 square feet of heating surface should be provided for each horse-power.

4. Fuel is wasted by brittleness, dust and small pieces dropping unburnt through the bars into the ash-pit.

5. The fuel is rendered less effective by the presence of earthy compounds, which form clinkers, abstract heat uselessly, and choke up the grate.

CHAPTER III.

FORMS OF STEAM BOILERS.

THE construction of a boiler should be regarded from two points of view. (1) There is the general form and structure adapted to support the bursting pressure of the steam. (2) There are considerations arising from the unequal action of the heat of the burning gases, and there are precautions to be taken for diminishing the waste of heat.

Considerations which influence the forms of boilers.—The early boilers were designed in simple defiance of all mechanical principles. Without doubt the safety of a boiler depends on the strength of the metal, but it is quite wrong to say that it is independent of the form of the shell, and any one who thinks for a moment on the subject will comprehend that a cylindrical tube of some kind is the proper vessel wherein to retain a supply of steam under pressure. The strongest form of a vessel for holding a gas under pressure is a sphere: that is the natural form for the purpose, as we learn in blowing a soap-bubble.

But no one would recommend a spherical

boiler, there are so many practical reasons against it.

The next best theoretical form is a cylinder, which may be of any size, and only suffers from weakness at two ends.

With cylindrical boilers there is a difficulty about strengthening the ends.

Theory would tell us to make the ends hemispherical, when their strength would be as much as twice that of the cylindrical sides. There is, however, a practical objection to forming a hemispherical or egg-end, as it is called, from the boiler plate; and the result is that the ends are commonly made of flat plate, which is strengthened artifically in such a manner as to give security against danger. As to the strength of boiler-plate the general rule is that a rolled plate is less strong per square inch of section than a thick bar of the same iron.

Also the reduction of strength is more marked in the transverse than in the longitudinal direction. There is a further subtraction from the strength by riveting; and according to Sir W. Fairbairn the breaking strains of riveted joints of boiler plate are estimated somewhat as follows:

	POUNDS.
Iron	50,000
Double riveted joints	35,000
Single riveted joints	28,000

Per square inch of section.

Since the plates after rolling are stronger in the direction in which they are rolled than transversely, it is apparent that they should be so disposed that the direction of greatest strength should encounter the greatest strain.

But the strain on the longitudinal section is greater than that on the transverse, and hence the plates, which may be 3 feet wide, are wrapped round the circumference, being laid in three lengths in order that the longitudinal seams may clear the brick-work seatings.

As a consequence of the theoretical disproportion between the two strains, *it is further recommended that the longitudinal seams should be double-riveted with ¾-inch rivets, pitched about 2½ inches longitudinally and 2 inches diagonally.*

The transverse seams are single-riveted, the pitch being 2 inches. To double-rivet them would appear to add but little to the strength of the boiler, though it would increase its weight and cost.

The tubes diminish the area of the flat ends and relieve the pressure tending to rupture the material on a transverse section. They further act as stays for holding the ends together.

On both these accounts a boiler with internal tubes becomes stronger than it would be without the tubes, so far as transverse rupture is concerned. The flat ends remain, however, a source of anxiety, and in order to support them gusset stays and tie rods are employed, as to which something will be said after the disturbing action of heat has been noticed.

EFFECT OF HEAT.

The wear and ultimate strength of a boiler are greatly influenced by adopting a construction which shall provide for the inevitable changes of form caused by unequal expansion and contraction due to changes in temperature.

Heat is motion, and as soon as the fire inside a furnace tube is lighted, the metal on the top becomes more heated than under the surface, and the tube arches itself in consequence of the greater expansion of the hotter portion.

And not only so, but the tube lengthens as a whole, and the flat ends bulge outwards. Finally the water becomes heated and the whole structure elongates, and unless sufficient allowance be made for the pulsating movements, straining will occur, which may possibly end in rupture.

The linear expansion of wrought iron (soft forged,) under the action of heat is stated to be .0012204 for a rise in temperature from 0° C., to 100° C.

Thus a bar of iron, 30 feet long, expands about $\frac{7}{10}$ inch for a rise in temperature of 132.2° C.

The expansion of the parts of a boiler as caused by heat is of course capable of accurate measurement; and in particular the so-called hogging of a boiler tube has been observed by applying three gauge rods attached at equal distances along the crown of the tube; each rod is carried vertically upwards, and passes through a stuffing box in the shell of the boiler, whereby it has been possible to observe very accurately the distortion of the tube.

One boiler experimented on was 28 feet long, and it was found that the tube rose $\frac{3}{8}$ inch when the flame passed around the boiler in the ordinary way along the side flues, and that it rose $\frac{1}{2}$ inch when the flame was carried directly into the chimney without heating the outer shell.

The gauge-rod at one-fourth the length of the boiler from the front end rose as much as that, and in one case $\frac{1}{16}$ in more.

Also the colder the water at starting the greater was the distortion, and generally the action was more severe just after the lighting of the fires. As soon as the whole of the water became permanently heated the gauge rods retired to their primary position, the distortion of the tubes seldom lasting for more than an hour. Mr Fletcher recommends that the end plates of boilers, to be used at a pressure of 75 pounds per square inch, should be $\frac{1}{2}$ inch in thickness, increasing to $\frac{9}{16}$ inch for increased pressure within moderate limits, excessive thickness being undesirable, as confining or restraining the necessary movement of the furnace tubes. The object is to strengthen the end plate; and yet to preserve its elasticity, and in carrying out this intention it is a rule to attach the plate at the front of the boiler to the shell by external angle iron. This mode of construction is not, however, adopted at the opposite end.

The furnace flues are a vulnerable part of the boiler, inasmuch as they are liable to yield by collapsing unless sufficiently strengthened.

The subject of strengthing the internal tubes of the internal fire flue boiler was investigated by Sir W. Fairbairn, whose experiments led to the following conclusions:

(1) The strength of a tube to resist collapse by external pressure is inversely as its diameter.

(2) The strength varies inversely as the length.

(3) The collapsing pressure in pounds per square inch

$$=806300\times\frac{(\text{Thickness of plate in inches.})^{2.19}}{\text{Length in feet}\times\text{diameter in inches.}}$$

In these experiments the ends of the tubes were firmly attached to rigid plates, and the vessel in which the compressing force was applied was a cast iron cylinder 8 feet long, 28 inches in diameter and 2 inches thick, which could be safely strained as far as 500 pounds per square inch.

Into this cylinder air was forced by a pump, and produced any required pressure on the surface of a quantity of water which nearly filled the cylinder, and in which the tube under trial was completely immersed. There were two gauges for reading the pressure, and a safety valve in addition, which was loaded by a weight.

The experiments made by Sir W. Fairbairn were valuable as calling attention to a material subject connected with the construction of boilers; but it appears that a special contrivance for strengthening furnace flues, while allowing at the same time for their expansion and contraction, had been originated some years prior to the researches to which reference has been made. At a meeting of the Institution of Mechanical Engineers held in 1876, Mr. Adamson stated that he had first employed a "flanged seam," as it is termed, for the strengthening of furnace flues, as early as the year 1851.

The Figs. 2, 3 and 4 exhibit three forms of joints as applicable to tubes subjected to external pressure.

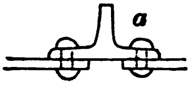


Fig. 2.

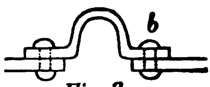


Fig. 3.

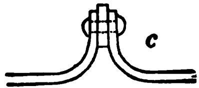


Fig. 4.

The first, marked *a*, consists of a ring of

T-iron, riveted as shown. It is abundantly strong, and is in a form which has been adopted for centuries past in strengthening guns. The weakness of a tube to resist a bursting pressure on a longitudinal section has been already demonstrated, and a common method of strengthening it has been to apply parallel rings at intervals along its length.

In this way steam cylinders of great length, which are subjected alternately to a bursting and compressing pressure, have been strengthened. Every one is aware of the accession of strength caused by the flange of a pipe. Since action and reaction are equal and opposite, we might have anticipated that a form of tube constructed so as to withstand a bursting pressure from within, would also be the form best adapted for resisting a collapsing pressure from without.

The difficulty of calculation in the latter case arises from the liability to deformation, which is soon set up, after which theory is of little use in enabling us to predict a result.

But although the joint *a* has ample strength, it is deficient in another quality which is of importance, viz., it does not permit any alteration of length. The whole furnace tube is rigid, and expands or contracts as one piece.

Whereas, in the joint marked *b*, the expansion or contraction of each length of the tube is provided for by the arched or the corrugated piece, and here there is increased strength combined with power of expanding or contracting freely. In the joint *c*, which is known as "Adamson's flanged joint," there is the strength of the T-iron directly combined with the curved end, which allows of unimpeded expansion or contraction.

The arrangement is most convenient and effective, and is particularly valuable as giving a seam where the rivets are protected from the furnace gases, and are, in fact, immersed in water, one consequence of the construction being that the joint will bear intense heat much better than any other where the rivets are exposed.

STRENGTH OF BOILERS.

While the pressure acts from the centre radially out to the circumference on an indefinite number of radial lines, the mathematics of the strength of the shell supposes that the pressure acts as the arrows *a*, *a*, *a*, *a*, in Fig. 6, that is tending to lift the top from the bottom. In Fig. 5 the pressure represented by arrow *a* is resisted by arrow *a'*, and similiarly arrow *b* and *b'* resist each other, and as each are equal, we may assume any one direction as that in which the pressure acts, as *a*, *a*, *a*, *a*, in Fig. 6, tending to part the boiler through the line *b*, *b*. We might, as stated above, assume any other direction of the pressure, but as the shell is as strong through *b*, *b* as it is through any other similar section, the strength as ascertained at *b*, *b* will be the strength of the boiler. The standard of comparison between different brands of boiler iron is based on the strength of a *square* bar of that iron, the sides of which are one inch in length, and a section through the bar representing one *square* inch. The tensile strength of such a bar will average 50,000 pounds. Assuming, then, that the pressure tends to tear the top from the bottom of the shell, it is apparent that the force acting to do this is represented by the length of the shell in inches, multiplied by the diameter in inches and by the pressure per square inch. The iron being the same thickness throughout the shell, we will also assume that a section one inch in length of the boiler is a *fac-simile* as to strength of any other section of similar length. The iron being $\frac{5}{16}$ of an inch thick and one inch (assumed) in length, it is plain that its strength is but $\frac{5}{16}$ of our standard of comparison, 50,000 (or 15,625 pounds). Now, as we have *two* sides of the boiler of $\frac{5}{16}$ thick, we must double the above amount, which = 31,250 pounds, as the strength of our section if it were without a joint. The strength of a single-riveted seam (Fig. 7) has been ascertained to be but 56% of the solid iron, and hence we must take that percentage of 31,250 to arrive at the strength of the weakest portion, which, of course, comes under the principle that the strength of any structure is equal only to its weakest part. 56% of 31,250 = 17,500, which is the product of the length of our section (1 inch) × by the diameter, say 48 inches, multiplied by the pressure per square inch necessary to burst the boiler. We have, therefore, in order to find the pressure per square inch necessary to burst the boiler, only to divide 17,500 by the diameter (the length being 1 inch or unity). This gives us nearly 365, which, divided again by our factor of safety, we will take as 6 (or the working pressure = to ⅙ the bursting pressure), and we obtain 60 nearly, as the proper working pressure, all of which tends to show that there are few steam-boilers in the United States that are not carrying more pressure than they ought to, or would be allowed to, if there were a law regulating the subject.

The strength of a joint, like Fig. 7, through the rivet holes, may be found by taking the length *a*, less the diameter of one hole multiplied by the number of holes. This leaves the length of solid iron between the holes, which, multiplied by the thickness and by the strength of a sectional inch, and from 15 to 20 per cent. deducted from the result, for injury done the iron by the punch, leaves the probable strength of metal between the holes. A staggered or double-riveted joint, shown in Fig. 14, has 70 per cent. of the strength of the solid sheet, as ascertained by Fairbairn's experiments. From inspection, it will be seen that there is as much metal between the holes *h*, *c* and *h*, *f* as there is between *c*, *f*. The metal between the holes is much greater than in Fig. 7, while the number of rivets is in proportion to the metal left between the holes, and hence the strength of the joint may be deducted from the metal left between the holes on a line, *a*, *b*. The strength of a rivet, to resist shearing, is about 50,000 lbs. per square inch, hence the sectional area of one rivet, multiplied by the number of rivets, and again multiplied by 50,000, will closely approximate their shearing or detensive strength. The rivet area strength should be kept as nearly equal as possible to the strength of sheet between the holes. The strength of the sheet to resist crushing is represented by Fig. 8. It is plain that if the rivet does not shear, nor the sheet rupture between the holes, the part *a* must be pushed or sheared out of the way, if rupture takes place. The resistance which the part or piece *a* offers is represented by the area *b*, *b*, or the length *d*, by the thickness of the plate plus the equal area *c*, *c*, and the whole multiplied by the detensive strength of the sheet, which is about 50,000 lbs. per square inch. In designing a joint, the piece *a*, Fig. 8, the shearing strength of the rivets and the strength between the holes should be as nearly equal as possible, to make as strong a joint as is possible. The joint shown in Fig. 12 has a welt piece *a*, added, and is much used in boiler work. It does not add as much strength as is commonly accredited to it. The line of rivets *d* and *f*, Fig. 9, are commonly spaced twice as far as the middle one *e*, and therefore present only half the shearing strength of rivets that line *e* does, and hence if rupture were to take place it would occur first through either *d* or *f*, if it were not for the stronger joint *e*. Therefore, after the internal pressure exceeds the strength of joint *d* or *f*, they

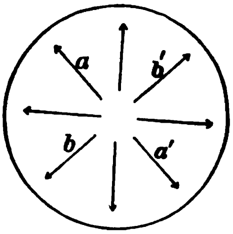


Fig. 5.

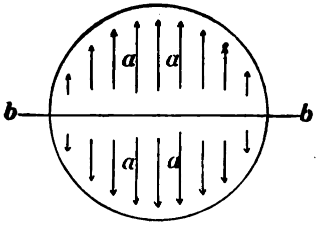


Fig. 6.

are no longer of use, as rupture is alone prevented by the superior strength of the joint *e*. The welt strengthens the joint simply because it puts the rivet *g*, Fig. 12, into double shear, that is, the rivet must be sheared through the line *h* and *i*, and therefore presents double resistance. The part of the plate in front of the rivet must be proportionally increased to meet the increased resistance, and the pitch of the holes increased as much as possible and still retain a tight joint. A combination of this joint and the staggered riveted joint would make the strongest possible longitudinal joint, as it would give the greatest amount of metal between the holes, and also present the rivet in double sheet. Fig. 10 shows a mode of making a joint by which the strength might be made equal to that of the solid sheet. The distance between the holes, by making the joint long enough, might be made equal to the width of the sheet, or in excess of it, for injury done in punching. The joint shown in Fig. 13 is not strengthened by the second row of rivets, barring the grip due to contraction of the extra

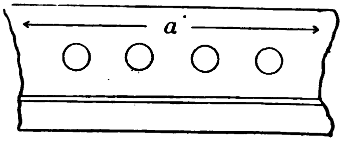


Fig. 7.

rivets, as the strength of metal between either row is the same, as is also the number of rivets,

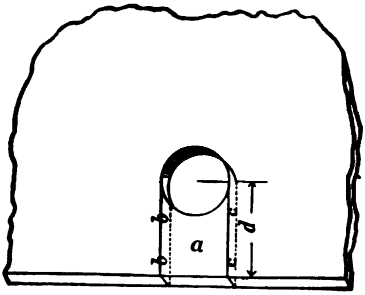


Fig. 8.

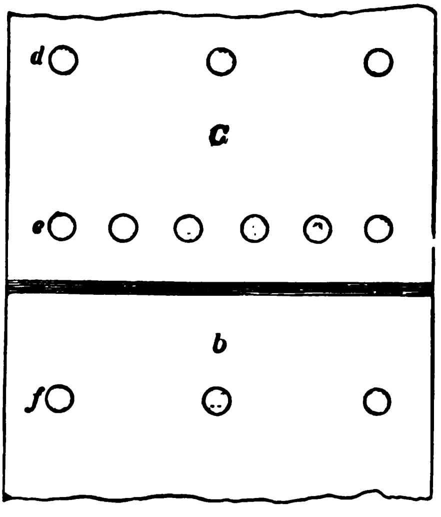


Fig. 9.

etc.; and the joint is as weak through one row of rivets as the other. We have been considering the longitudinal strength only. The transverse strength of a boiler is that which prevents one end from being torn from the other. This strength is represented by the thickness of the iron × by the circumference × by the strength of a sectional inch, and the whole less the percentage of loss due to the kind of joint. However, a single rivet joint will suffice, generally, as the boiler is about twice as strong transversely as longitudinally.

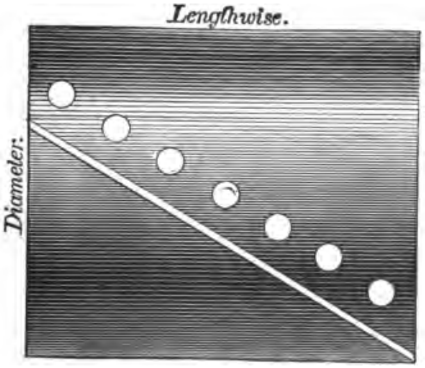


Fig. 10.

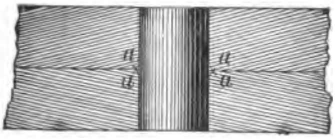


Fig. 11.

Persons buying boilers would do well to have samples of the iron, from which the boiler is to be built, tested for tensile strength, and take no interested parties' word, or bond either, on that point. An actual test settles the matter beyond doubt. All holes should be *drilled* and rounded in the inner edge, as shown at *a*, *a*, *a*, *a*, Fig. 11, as a drilled hole, if not rounded, will shear the rivet sooner than a punched hole. Flat surfaces should be stayed to from six to ten times the pressure they are to resist. The *area* of rivets holding crow feet to boiler heads, etc., should be equal to the stay coupled to said crow foot. It is a good plan to make stays coupled to crow feet slightly shorter than the required length. Expand them by heat, and, on cooling, they will all be found tight and necessarily bearing their proportional strain. It is a *bad* plan to collect one end of stays (such as crown sheet stays when stayed to shell) into a smaller area than the other, as it is plain to be seen that the smaller area must resist more than its share in proportion as its area is smaller.

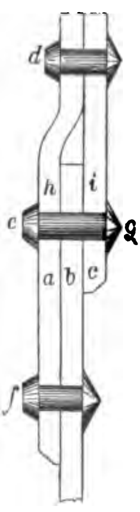


Fig. 12.

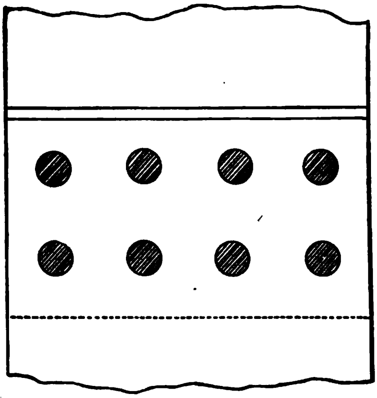


Fig. 13.

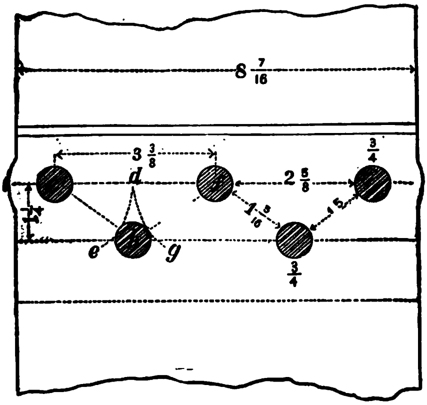


Fig. 14.

CHAPTER IV.

BOILER ECONOMY.

THE scientific measure of the economy of a boiler is the amount of water evaporated into saturated steam, at some unit of evaporation per pound of combustible. This unit, for convenience of comparison, is usually 965.6, or, the units of heat required for the vaporization of one pound of water, under a pressure of one atmosphere. However convenient this may be, it hardly ever expresses the amount of water which would be evaporated if the evaporation were to take place under one atmosphere, instead of being reduced to that condition from some other pressure at which it actually did occur. Neither is it a correct expression of the relative economy of different boilers, and particularly of different types of boilers, unless the evaporation takes place under the same pressure in each instance, and unless the pressure is the one at which the boiler is to be used. Manifestly a boiler designed to economically generate steam under a pressure of twenty atmospheres would not show equal relative economy when working under a pressure of two or three atmospheres, nor would the reduction of the performance of two boilers, working under these pressures respectively to some standard unit, be in any sense a fair criterion by which to determine their relative economy.

The practical measure of the economy of a boiler in any event embraces more than this, it being necessary to take into account two, and sometimes all of the following considerations, viz., the amount of water evaporated into dry steam of the required pressure, the amount of space occupied by the boiler and its appliances, the probable necessity of stopping for repairs, and the cost, in which is properly included the expense of keeping in repair, renewing, etc. As an instance of the application of the foregoing, it would be useless to demonstrate that a boiler three times as large would increase the general economy of the locomotive engine, looked at in the light of the fuel consumed, because the economy in that case consists in getting a large amount of work from a very small boiler.

The space occupied and the steam generated take precedence, and must be first considered. On the other hand it would be an example of bad engineering, under ordinary circumstances, to place for permanent stationary purposes an ordinary locomotive boiler, and then by forced draught and unceasing attention get the same amount of work from it as in the case of the locomotive engine. It may thus be readily seen that boiler economy under different conditions is the product of widely different factors. Theoretically, the subject has been widely speculated upon. Practically, however, it is always worth the consideration of the party who pays for the coal. One pound of good anthracite coal has a combustible efficiency of 15,000 heat units. Hence, if this could all be utilized it would evaporate under a gauge pressure of 75 pounds, and from a temperature of 60°, 13+ pounds of water. The average performance of boilers is hardly if any more than ½ of this. A consideration of what becomes of the unutilized heat is perhaps the best way to determine how to effect a saving. In the first place, it is not all wasted. *Man cannot change the laws of nature.* At the best he can only tack his little endeavors to these laws, and get them carried along as far as possible, and then submit to seeing them dropped. As in the cylinder of an engine the useful effect is limited to a difference of temperature expressed in a few degrees, so in the boiler, the useful effect must be obtained between the temperature of the fire, which may be termed the initial temperature, and that of the escaping gases, or the terminal temperature. Fortunately, these extremes—initial and terminal temperatures—may be, and usually are, much more widely separated in the case of the boiler than in the cylinder of the engine. Different temperatures in contact seek to equalize themselves. If we take into consideration the inside of the boiler where useful work is being done, and the outside where a contrary condition exists, it is evident that however perfect our appliances for utilizing the heat, the simple fact that we are working for a temperature higher than that which we are working in, is proof that the result will not in any event be absolutely perfect. This lack, however, is not a loss, since it is something we never had nor can have.

This consideration of the subject is mainly interesting as indicating the sources of greatest loss, and hence the direction in which to look for a saving. Assuming, as we have previously done, that one pound of good coal will, under perfect conditions, evaporate 13 pounds of water, how much ought it to do in practice? Ordinarily, and where space has but little to do with the subject, seventy per cent. of that amount, or nine pounds. Where the heat from a high pressure engine is available, this amount should be increased to not less than ten pounds. Whenever, then, less than nine pounds of water without a heater, or ten pounds with a heater, are evaporated with one pound of good coal, and under a gauge pressure of 75 pounds, it is safe to assume that better results may be obtained. But if nine pounds ought to be realized, and no more (at an average) than 6½ pounds are realized, what has become of the heat represented by this difference? The heat amounts to 28 per cent. of all the fuel burned, or the consumption of forty per cent. more coal than is necessary. It has been lost mainly in radiation from the outside of the boiler or brick work, and passed off with the products of combustion. It represents a total loss, because the possibility of saving it exists. Regarding the first mentioned source of loss, external radiation, the remedy is obvious—more careful attention to outside protection. Regarding the second source, the escape of heat past the heating surfaces of the boiler, more attention may be necessary to make plain the causes of the loss occurring here.

The horse-power of a boiler is, unfortunately, an almost meaningless expression. If we assume it to be the evaporation of 30 lbs. of water at a pressure of 75 lbs. (per gauge), it may be undoubtedly done on the more usual allowance of 15 square feet of properly proportioned heating surface. Assuming that 15 feet may be sufficient, it does not follow that because a certain boiler has fifty times 15 feet of heating surface it is a fifty horse-power boiler, any more than it does that a definite cylinder capacity makes a certain horse-power engine without reference to piston speed, steam pressure, or, whether the construction is such as to allow a proper use to be made of its capacity.

Again, assuming the 15 feet of heating surface to be ample when the boiler is new and clean in all its parts, we have yet to meet the condition of its becoming old, and its efficiency impaired by the formation of scale. It has been demonstrated that this may result in loss of efficiency anywhere to the extent of 50 per cent., according to the nature and thickness of the deposit. It is useless to say that scale should not be allowed to form in a boiler, because it *does* in spite of all known precautions. This may not always mean the impairment of economy to that extent, but it *may* mean its impairment in even a greater ratio. The presence of scale on the heating surfaces of a boiler does not imply that the heat will not penetrate it, and so become absorbed by the water in contact, but it will not penetrate near so readily as it would the iron alone; hence in effect it has lessened the heating surface, since the effective heating surface should be measured from a consideration of its quantity and quality.

If the economy is to be preserved the work done must be diminished, when by keeping the heat longer in contact with the heating surfaces more time is allowed it in which to pass through them. The reverse of this is what is usually done; that is, the boiler is kept up to its work by forcing a greater quantity of heat and, of course, at an increased velocity, by the heating surfaces, and the result is the escape of a larger percentage into the chimney. But this is not all nor perhaps the worst of it. The heating surface to be effective must have water in contact with it. The pepper box, or as many tubes as you can get in style of construction, has the serious fault of not having sufficient room for the escape of steam and the circulation of water. This defect is further aggravated by filling up a portion of the scanty room by a rough, uneven deposit, over and through which the steam and water are hindered from passing; so that instead of having water (which absorbs heat with remarkable rapidity) at all times in contact with the metal, or at least with the scale with which it is covered, the place of the water is partially occupied by steam which absorbs heat much less readily. Any one who has seen this class of boilers "raise" the water 6″ when working over a hot fire, often to such an extent that it becomes absolutely necessary to open the fire-door for safety, will have no difficulty in believing that the space which ought to be occupied by water frequently has very little water in it. Thus the efficiency of the boiler has been curtailed by providing a slower conductor of heat, and, at the same time providing a power absorber. If to this condition is added what is almost universally the fact, viz., that the boiler, scarcely large enough when new, has had additional demands made upon it from time to time, we shall see one cause for the low evaporative duty of a great many boilers, and perhaps one serious cause for accidents as well. Since the boiler will be new but once while it will be old as long as it lasts, it should be calculated for good duty when from the causes named it shall become less effective. The result will be increased economy in the consumption of coal in the beginning, gradually subsiding to good standard economy, and an increased economy in the way of repairs which will much more than overbalance the increased first cost.

As to the kind of boiler, almost any will give good results which has sufficient run to allow time for the transmission of heat, with water at all times in position to absorb the heat transmitted. Upon general principles, that boiler will show the best results which presents the greatest amount of *efficient* heating surface, with the least amount of external radiating surface.

This statement, however, must only be taken in a general way. A great many patent boilers are devised, evidently with this object in view, but at least nine-tenths of them make better failures than anything else. The locomotive boiler, as used in locomotive practive, is economical only so far as space occupied is concerned, because it is used beyond its normal capacity. The same type of boiler, used for stationary purposes, at a moderate rate of combustion, say from seven to eight pounds of coal per hour on each foot of grate surface, and with the connection with the up-take *below* the tube sheet, gives results hard to beat. Of course, the exposed surface should be thoroughly clothed to prevent radiation. Objection is sometimes made that this boiler requires closer attention as to water than some others, but this is hardly worth considering in the light of an objection. Neither are they particularly liable (as is often asserted) to failure in their fire boxes when properly constructed. With extremely bad water or very poor coal, their use is not commendable. Their first cost is not much more than the return tubular (including brick work), while the cost of maintenance is about the same.

The greater loss of heat in radiation from masonry will generally render the return tubuar scarcely as economical as the locomotive boiler, but the greater simplicity of the former will, perhaps, generally overbalance this loss. In extreme cases of bad water, or in out-of-the-way places, in the event of repairs being necessary, the single, or two-flue boiler, or even the plain shell with no flues, will be used; not with the expectation, however, of obtaining high economic results. Whenever, as in the majority of instances, the economy consists in evaporating the greatest amount of water with the least amount of coal, the result will be reached by providing large (long) heating surface, slow combustion, and ample room for the escape of steam and the circulation of water. Undoubtedly, this means increased first cost, and if it cannot be shown to pay, is not worth consideration. Fortunately, the means employed to insure economy in the use of coal are the very means that should be employed to bring about the same result in regard to repairs, safety, and renewing.

CHAPTER V.

ECONOMY IN THE USE OF STEAM.

ECONOMY in the use of steam requires that it shall be admitted to the cylinder at or near the beginning of the stroke of the piston, at a pressure not much below that in the boiler; that this pressure shall be maintained practically constant for a portion of the stroke, and that then the supply shall be cut off quickly, the remaining portion of the stroke being completed under the expansion of steam in the cylinder. Under these conditions the pressure of steam in the cylinder will be so far reduced at the conclusion of the stroke that little or no more useful work can be performed by it.

Then this steam must be exhausted so freely that back pressure in the cylinder will be reduced to a minimum.

Whatever tends to prevent in any degree the attainment of these ends, to that extent reduces the gain otherwise attainable by working steam expansively, and brings about an important loss due to the effort required to expel the steam from the cylinder after it has done its work.

To secure these ends, which underlie economy in the use of steam in the steam-engine, it is imperative that the ports shall be of suitable size; that the steam port shall be quickly opened for the admission of steam; that ample opening shall be maintained up to the point of cut off, and that then (the piston being in rapid motion) the port shall be so nearly *instantaneously* closed as practically to avoid what is technically termed wire-drawing at cut off.

It is also equally important that the exhaust valve shall open a free passage for the escape of the steam just previous to the termination of the stroke; that this amplitude of opening shall be maintained until near the termination of the return stroke, and that then the exhaust closure shall be reasonably rapid, securing the advantages due to compressing the imprisoned steam.

The important ends of rapid valve opening and closure, and the maintenance of ample effective port area, are accomplished by the *wrist plate* motion peculiar to the Corliss valve-gear, and by the form of valve employed *in a way seldom successfully accomplished by other means;* the location and construction of the valves is such that there is no difficulty in securing the long and large ports, which are essential.

At the same time, *clearance*—a necessary evil, always representing loss—is reduced to a minimum. Numerous attempts have been made, some of them very ingeneous, to devise a valve-gear that should perform its functions as well as this, but they have generally failed. For reasons, amongst others that have been previously named, the Corliss type of engine has for a quarter of a century been used for illustrating economy in the use of steam.

BALANCED VALVES.

The attempt to construct a balanced steam-valve that should not waste steam by leakage, or be otherwise unsatisfactory, has nearly always failed.

But in many instances the use of such a valve, with its parts and pieces subject to derangement, is a necessity, and its objectionable features must be taken as a matter of course. This is the case when the governor, in its efforts to control the speed, must move a heavy valve: It is also generally true in the instance of separate expansion valves working on the back of a main slide, because otherwise the pressure will be so excessive as to rapidly destroy the valve and seat, to say nothing of severely taxing the parts through which motion is transmitted to the valve. *No such necessity exists* in the use of the Corliss type of valve. The ease with which these valves work may be demonstrated by observing how readily they are moved by hand while under full steam pressure.

REGULATION.

The economy of close regulation of speed is much better understood than it formerly was.

In many industries a carefully determined speed of machinery cannot be exceeded without material, sometimes very serious loss, while if the speed is slower the production is correspondingly decreased, and frequently the product is inferior in quality.

If the speed of a steam-engine cannot be controlled within narrow limits, then the normal speed must, as a matter of course, be considerably slower then that at which the best results will obtain, in order that the maximum speed may not exceed the limit.

When the regulation is close the normal speed may be almost exactly the maximum, instead of several per cent. slower. In one case the machinery is kept in motion at a speed representing its greatest productive capacity, instead of as in the other, several per cent. below this.

A closely regulated engine may readily increase the capacity of machinery from *five to ten per cent.*, as compared with one in which the regulation is only moderately good; a matter of no small importance in large manufacturing interests.

We believe this is a subject which manufacturers may profitably investigate further than they have yet done.

The steam-engine governor is extremely sensitive to the effect of irregular resistance, or work, and when this is exacted of it, it must be at the expense of good regulation.

The duty of the governor should be to indicate the point of cut-off, not to accomplish the work of cutting off.

FOUNDATION.

It is essential that a steam-engine be placed upon a solid, unyielding foundation.

It is unwise to pay for a good machine and then set it upon a foundation that will yield to the weight and working strains in such a way as to throw the parts out of line, bringing about injurious wear and derangement. Large cut stones make the most solid foundation, but these are costly; hard-burnt brick make an excellent substitute.

These should be laid in cement, with their joints.

No stones are required except the cope-stones, which should be cemented to the brick work. This is assuming that the bottom rests on solid ground.

If this is not reached at the depth indicated, the excavation should be deeper, the bottom rammed hard and filled up to the point at which the foundation proper is to begin with rough stones laid in cement. The foundation should be completed at least fifteen days before the engine is placed on it, by which time, if properly laid, the brick-work will be practically one mass.

CONDENSING ENGINES.

When there is a supply of water without cost, a condensing engine will, under favorable conditions, be more economical than one working non-condensing. If city water must be paid for, it is generally advisable to run non-condensing.

To get the advantages of a condenser, the engine should be well loaded. The saving in fuel should not be less than twenty per cent.

If it is necessary to run with low steam pressure, a condenser will be found especially advantageous.

The amount of condensing water required will depend upon its temperature and upon other conditions, about seventy gallons per horse power per hour being a fair average.

For large power the COMPOUND ENGINE is unquestionably the most economical in the use of steam, and since the principles involved in its operation have come to be better understood, and approved construction therewith adopted, it has correspondingly grown in public favor.

As is well known, there is a comparatively early limit to expansion in a single cylinder, owing to the fact that condensation and subsequent reëvaporation will by robbing the cylinder of heat, more than offset the gain from increased expansion if carried beyond this limit.

The extent of condensation in a given cylinder, under given conditions, otherwise, is mainly controlled by the variation in temperature, as between that of the steam as it enters the cylinder from the boiler on one hand, and the temperatures during and at the end of expansion and during exhaust on the other.

In a single engine (so called to distinguish it from the compound) this difference will all occur in one cylinder, and when high rates of expansion are employed, the steam condensed is often in excess of that usefully employed. In a compound engine these unavoidable extremes of temperature are divided between two cylinders in such a way as to greatly modify the effect.

This permits expansion to be carried to an extent in a compound engine that would never seriously be thought of in the instance of a simple engine, effecting a very material saving in fuel.

The compound engine has also some other advantages over the single engine, such as a better distribution of strains and a more uniform rotative effect on the crank-pin, and hence on the shaft.

For a given power the first cost of a compound engine exceeds, of course, that of a single engine;

but this is partially balanced by the saving in boiler capacity, and is very soon saved in economy of fuel.

There are different plans of compounding, one of which is to use, in effect, two engines connected to cranks on the ends of the shaft.

This has the advantage of occupying but little space compared with the two engines side by side, and does not present so many moving parts to be cared for.

It also has the advantage due to the cylinders being placed close together, so that the certain loss due to passing the steam a considerable distance, from the high to the low pressure cylinder, is avoided.

This last consideration should always exert due influence when arranging to use a compound engine.

In the gear ordinarily employed with the Corliss engine, the construction is such as to lead to some inconvenience and annoyance from the parts wearing loose and becoming shaky.

The effect of the action of the releasing gear is also to slightly disturb the governor; the latter is the more serious fault, because its tendency is to impair the regulation, preventing its being brought to that degree of perfection which is desirable.

CALCULATION OF WORK DONE IN A COMPOUND ENGINE.

In the working of a compound engine, where the small cylinder exhausts into the large one, the work done in a stroke depends on the size of the large cylinder, and is the same as that which would be performed in a single cylinder of the same content, by expanding to the same extent from a like initial pressure.

This proposition is easily proved, and any one may satisfy himself that it is approximately true by examining a well-formed indicator diagram as taken from a compound engine.

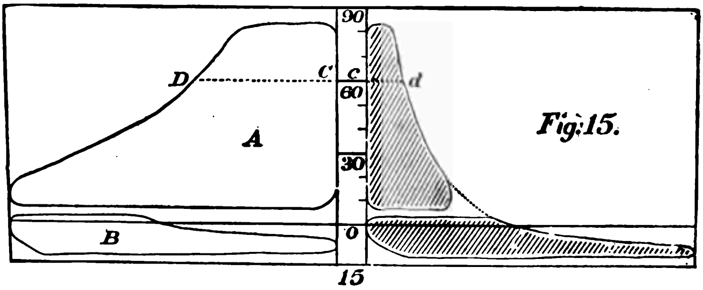


Fig. 15.

Referring to Fig. 15, which is from an engine having a high-pressure cylinder 18 inches in diameter, with a 6 feet stroke, and a low-pressure cylinder of 36 inches in diameter, with a stroke also of 6 feet, the number of revolutions being 34 per minute.

Since the lengths of stroke are the same, and the areas of the pistons are as 1 to 4, it follows that the indicator diagram marked *A*, as taken from the high-pressure cylinder, would be reduced to the same scale as that from the low-pressure cylinder marked *B*, if we supposed the diameter of the latter cylinder to be 36 inches and the stroke $\frac{6}{4}$ feet, or one-fourth of that which it really is.

This result is set out in Fig. 15.

The diagram marked *A* is reversed in position and repeated on the right hand side by measuring off a series of horizontal lines, such as *c d*, and making *c d* equal ¼ *C D* in every case.

In this way the upper shaded area represents the work done in the high-pressure cylinder as it would appear on the scale adopted in the low-pressure cylinder. The bottom shaded area is merely a repetition of the area *B*.

When the two diagrams are put together it will be seen that the two portions of the expansion curves fit very fairly or run into one, and that the expansion commenced above *d* is carried on throughout the stroke.

It will be noticed that there is a little want of similarity between this diagram and Fig. 16. Here the steam line in *b* is horizontal at first, and then slopes downwards.

That it is horizontal at all is owing to some peculiarity in passing the steam from one cylinder to the other, as there should be the slope of an expansion curve throughout. But any deviation from theoretical proportions does not affect the general inference to be drawn from the two diagrams when viewed together, and we see that the expansion which has occurred in the high-pressure cylinder might very well have taken place in the low-pressure cylinder, as something which preceded the actual expansion therein.

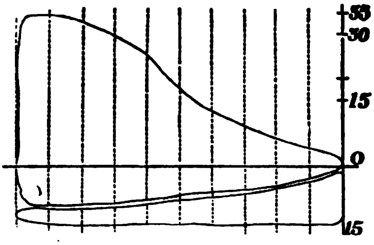


Fig. 16.

ENGINES WITH CRANKS AT RIGHT ANGLES.

For many purposes it is enough to have an engine with a single steam cylinder, or an equivalent engine, with a pair of cylinders acting as one only, but on the other hand, there are numerous instances where two engines should be placed side by side and work cranks at right angles to each other.

This is particularly the case in applying steam-power to flour mills or to cotton mills, where it is of consequence to preserve the rotative pressure on the crank as nearly uniform as possible, and to maintain a smooth and even motion, or again in marine engines, for convenience of starting in any position, the same rule would hold, and before proceeding further it may be useful to point out the reason for the greater uniformity of rotative pressure which is a consequence of working with a pair of cranks at right angles.

That the variations of tangential pressure on the crank of a direct-acting engine are represented by the vertical lines on a diagram similar to that shown by the dotted curve *b f* in Fig. 17.

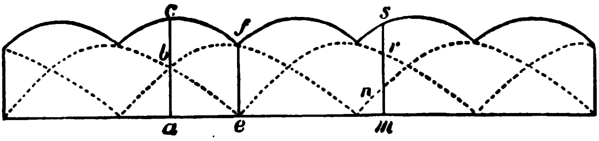


Fig. 17.

Putting a series of such curves end to end, we obtain a graphical indication of the fluctuations of tangential pressure during the working of an engine with one cylinder.

The force is zero at dead point, and rises to *f e*, its greatest value, after which it sinks again to zero.

But if there be a pair of cranks at right angles, a second series of diagrams of rotative pressures must be superposed upon the first series, as shown by the second set of dotted curves, whereof one portion is marked $b\ e$, and the final result is exhibited by the upper line, not dotted, which is obtained by adding together the pairs of ordinates at each point. For example:

$$\begin{aligned}&NM+MR=MS.\\&AB+AB=AC.\\&EF+O=EF.\end{aligned}$$

The greater uniformity of rotative force is apparent, and it would be improved by cutting off at half stroke in each cylinder, for then the curve $b\ e$ would be hollowed out and reduced, while the part $b\ f$ would be unaffected, and the upper resultant wavy line would become more nearly horizontal.

By proceeding in this manner it is easy to set out a diagram of the rotative pressure upon the cranks of any pair of engines working under given conditions.

In applying these principles to direct-acting engines, where two cranks at right angles are to be connected with the cylinders, there are different methods for adoption, each of which has its advocates. One plan very commonly met with has been to place the high and low pressure cylinders in pairs, with their axes in the same straight line, so that one piston rod serves for both. Thus, in marine engines, with the cylinders vertical, there may be—

(1) The high-pressure cylinder above the low-pressure cylinder.

(2) The low-pressure cylinder at the top.

(3) The low-pressure cylinder encasing the high-pressure cylinder.

But in each of these cases, as also in compound horizontal engines, it is usual to confine the expansion to one pair of cylinders.

THE USE OF AN INTERMEDIATE RECEIVER.

In another class of compound engines there are two cranks at right angles, but only one cylinder connected with each crank.

Here each cylinder forms, as it were, an engine complete in itself; the cylinders (called A and B, as before) are placed side by side, and are of equal length, and the point to be noticed is, that the pistons in A and B no longer move together, but that one leads the other by half a stroke.

It is clear that the mode of exhausting at once from A into B is no longer applicable, and that some special method of distributing the steam, different from anything that we have yet seen, must be arranged.

The difficulty arises from the fact that the directions of motion of the pistons cross each other, whereby, for example, when the piston in A is at the end of its stroke and about to ascend, that in B is in its middle position and is desending.

In order to get over this obstacle, Mr. Cowper has proposed to place an intermediate receiver between the cylinders *A* and *B*, which shall act as an exhaust reservoir for the steam coming from *A*, and as a boiler for the steam going into *B*. It appears that engines with a receiver have worked well in practice, but it seems difficult to justify the use of this arrangement by a strict reference to the principle of the theory of heat.

A general idea of the arrangement of the engine proposed by Mr. Cowper may be gathered from the Fig. 18, where the cylinders *A* and *B* are placed side by side, and the high-pressure cylinder *A* is enveloped in a steam receiver or reservoir, marked *C*, the content of which is perhaps three times that of *A*.

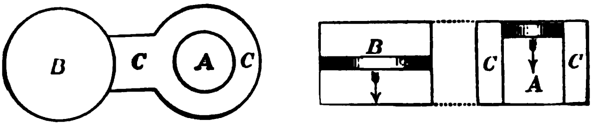


Fig. 18.

In a working engine on this plan, steam (say at 70 lbs pressure) would enter *A* and be cut off at half-stroke; it would thus expand and finally exhaust itself into the receiver, where the pressure would vary from, say, 10 lbs. to 14 lbs.

The receiver would supply steam for the low-pressure cylinder *B*, just as if it were the boiler of an ordinary engine, and the pressure of the steam in *C* would fall to 10 lbs. when the demand upon it was made, but would rise to 14 lbs. when fresh steam entered it from *A*. The temperature of the steam in the jacket surrounding *A* is therefore much below that of the entering steam, which is so far a departure from the old practice.

CHAPTER VI

SLIDE VALVES.

THIS is a branch of the subject deserving the special attention of the engineer.

Its importance in regard to the economical working of the steam engine cannot be overestimated.

The slide valve ordinarily used in high-pressure engines, and the manner of its operation, are well known to nearly every practical mechanic and engineer. It will be remembered that the operations of admitting the fresh steam, and releasing, the waste steam, are alternately performed by the same valve and the same motion.

The valve being made to slide backwards and forwards upon the face of the ports, opens and closes the several passages in their turn.

The two extreme ones, called the steam ports, communicate with each end of the cylinder.

The middle one is called the exhaust port, and its corresponding passage terminates in a pipe open to the atmosphere.

Steam is admitted freely into the steam chest from the boiler, and the valve is made of sufficient length to cover all the ports, when it is placed in the centre of the stroke.

When it is in this position no steam can enter the cylinder, but as the valve moves on one of the ports opens, the arrangement of the valve gearing being such that when the piston is ready to begin its stroke, the steam port begins to open.

During the stroke of the piston, the valve not only travels to the end of its stroke, but also returns to the point from whence it set out, and its continued motion in the same direction finally closes the port and prevents any further admission of steam. The steam has now done its work, and must be removed. In the middle of the valve a hollow chamber is formed of sufficient length to open between the ports.

As soon as the edge of this chamber passes the edge of the steam port, the pent up steam finds vent and rushes through the exhaust port and escapes through the exhaust pipe into the atmosphere.

Now looking at Fig. 19 you notice that

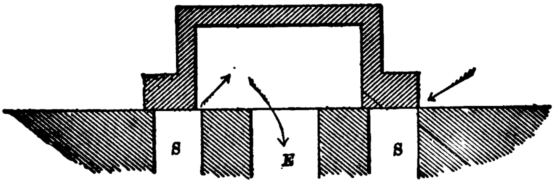


Fig. 19.

the exhaust port opens when the steam port closes, and that both happen just at the end of the stroke.

The perfection of a steam valve, other things being equal, consists in the degree of nicety with which its motion is timed relatively to the motion of the piston.

The functions of the piston are absolutely dependent upon the proper timing of the admission and release of the steam. A very slight and apparently trifling error in the adjustment produces a most serious effect upon the consumption of fuel. If from any cause the valve should open to admit steam for a fresh stroke before the preceeding stroke is finished, it opens too soon, and an unnecessary resistance to the piston is produced.

If on the other hand, the valve should delay its opening until the piston had begun its return, it opens too late, because thus the steam has to fill uselessly the space left vacant, and hence a waste of steam, and a loss of power.

As far then as the admission of steam is concerned, it is a necessary condition that the *steam* ports should open neither before nor after, but at the precise moment when the stroke commences. Some engineers recommend giving the valve "lead" as it is termed, that is to say, setting it so as to open a little before the end of the stroke, but it is an open question whether the slightest advantage is gained by so doing to a greater extent than is necessary to compensate for any slackness or lost motion in the valve gearing, or for their expansion when heated by the steam, and $\frac{1}{16}$ of an inch is quite sufficient in a well constructed engine.

It is also an open question whether it would not be better to bring the piston to a state of rest by the "compression" of the exhaust steam than by means of any lead to the steam valve at all.

Now the valve shown in Fig. 19, ratifies the conditions for the *admission* of the steam: it opens exactly at the right time, the steam begins to enter as the piston begins to move, as it follows it steadily and effectively throughout the stroke. Whatever *time* the piston takes for its journey, the steam is allowed as much time to follow it. At first the opening is small, but then the motion of the piston is comparatively slow, and therefore the supply keeps pace with the demand.

As respects the *release* of the steam when the stroke has been completed, the performance of this valve is altogether unsatisfactory, and here lurks the cause of the difference in the performance of the old and the later engines. But it may be said that the release does appear to take place at the right time, because it occurs just when the piston has finished its stroke, and if it were to occur before, a loss of power would ensue.

This is a very plausible view of the case, and the one which delayed for years the saving of fuel which has since been effected.

Sufficient attention was not bestowed upon what was going on in the cylinder, or upon the facts which might have indicated them; to fill and empty a cylinder full of steam are operations requiring *time*.

The time required for filling the cylinder with steam necessarily corresponds with the duration of the stroke, whatever its duration may be. But this cannot be the case as regards the second operation—emptying of the cylinder.

This ought to be performed in an *instant*, or otherwise the steam continues pent up when it ought to be liberated, when it ought to assume its minimum pressure—the pressure of the atmosphere—and exerts an injurious counter-pressure against the piston, tending to increase the resistance to be overcome. To effect the free and rapid discharge it is necessary not merely to open the communication to the exhaust pipe, but to open a wide passage and to have this done by the time the piston commences the return stroke.

The valve alluded to cannot accomplish this, its motion being gradual, not instantaneous, as it should be.

The passage only begins to open when the piston is in turn, and it is not wide open until the piston has travelled through one-tenth of its entire stroke. The steam in the cylinder is restrained from escaping, being, as it were, wiredrawn in the passage out, and consequently takes considerable time to assume the pressure of the atmosphere. In the meanwhile the new stroke has begun and been partly completed, and so far the piston has had to contend with a resistance altogether illegitimate, a resistence in many cases—especially at high speeds —nearly equal to all the other resistances put together. It was not until the year 1838 that the true cause of the trouble was suspected, and a remedy applied.

It had been thought before that time, that giving an engine lead tended to improve its speed when already running at a high rate.

This was attributed to the opening of the steam port being wide at the commencement of the stroke, thereby increasing the facility for the entrance of the steam in following up the piston.

Its true explanation was found to be the *earlier release of the waste steam* and consequent diminution of resistance. As sometimes three-eighths of an inch, or even one-half of an inch "lead" was given to high-speed engines, it was decided to try the effect of opening the exhaust passage earlier by the same amount, while the steam port should still be made to open only at the beginning of the stroke.

An engine was chosen for the experiment the valve of which resembled Fig. 20, placing the valve on the ports so as to allow the exhaust passage to be three-eighths of an inch open, the steam port at the same time to be one-quarter of an inch open. This space, therefore, was closed by adding to the length of the valve at each end one-quarter of an inch.

The eccentric was, of course, shifted on the shaft to correspond with the alteration, and the engine with the altered valve (see Fig. 20) again set to work.

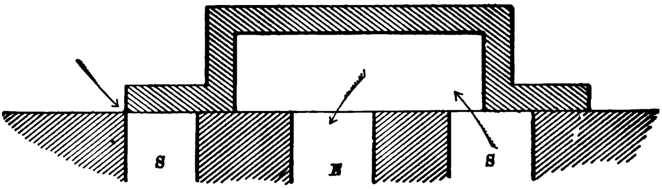


Fig. 20.

It is almost useless to say the saving of fuel was very great. The amount by which the valve at each end overlaps the steam ports (see Fig. 20) when placed exactly over them is technically termed the "lap." The lap of the valve being three-eighths of an inch, the exhaust passage was about three-eighths of an inch open when the stroke was finished.

LAP.

The importance of putting lap upon a slide valve will be better understood by noting what would happen without it.

If there were no lap the opening for steam would be represented by *O D* (see Fig. 21), and the result would be that the steam port could only be perfectly closed at the precise instant when the valve was in the middle of its stroke, and moving rapidly. From what has been said it is apparent that a valve of this kind (one without lap) is unsuitable for an engine, the better plan being that the steam should be compressed or cushioned on one side of the piston, so as to assist in bringing it to a state of rest, and that the driving pressure on the opposite side should be relieved by opening a passage to the exhaust or releasing the steam just before the stroke terminates.

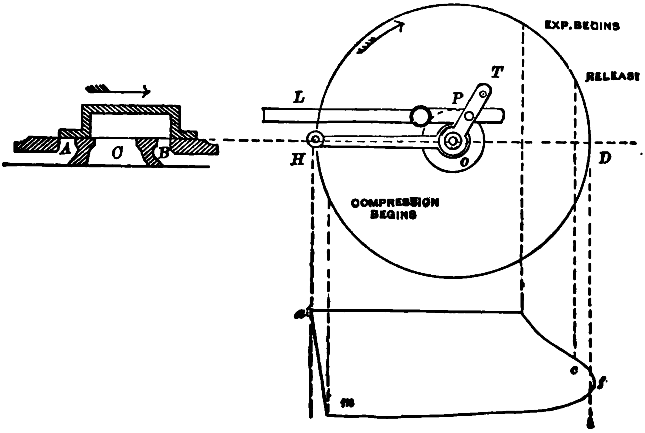


Fig. 21.

This prevents the violent jerk and strain which would come upon the crank-pin if the steam were thrown with full force upon the piston when the crank is on the centre. The value of an indicator diagram in interpreting the action of a slide valve can now be made clear.

Referring to Fig. 21 it will be noticed that the moving parts are attached to a board carrying a sheet of paper on which the circles describing the centres of the crank-pin and centre of eccentric are recorded, and below this is a space for tracing the indicator diagram.

The crank and connecting rod which actuate the piston are at the back, but an index arm *O H* is placed in front, and moves with the crank, thereby transferring its apparent motion to the part where it can be seen. The eccentric is represented by an actual crank, *O P*, whose extreme end describes the smaller circle, and the rod *P L* carries on the motion of the valve.

The point *P* can be shifted along the arm *O T*, thereby varying the amount of travel of the slide, and the length of the rod *P L* can also be adjusted. In this way the effect produced by any deviation from the proper length of the eccentric rod can be studied. We are now prepared to trace out the diagram as given by an indicator.

The crank being horizontal with the piston at the end of its stroke, the first thing to be done is to place the valve in the correct position for admitting steam, by setting back *O P* until the lap is allowed for.

The valve then opens, and if the pressure of the steam be sufficiently maintained the indicator pencil will trace the line *A B*.

When the crank gets to the end of the first dotted line, *A* is closed, expansion begins, the pressure falls, and we have the expansion *B C*.

At the point marked "release" the valve is moved so far to the left as to open a passage from *A* to *C*, and the release of the steam (exhaust) begins. The pressure falls from *C* to *F*, and continues very low till the point marked "compression" is reached, when *B* closes, and the steam in the corresponding end of the cylinder is "cushioned" so as to increase its pressure, the pencil rising from *M* to *A* when the double stroke has been completed. The object and effect of putting "lap" upon a valve are twofold:

1st, to give a free release to the exhaust steam, and 2d, to produce a fixed amount of expansion.

The "lead," of which mention has been made, is outside lead, that is, it relates to the *admission* of the steam; but of course, "lead" can be given to the exhaust side of the valve, in which case it is called *inside lead.*

The four principal points in the valve motion are: 1st, the admission of the steam; 2d, the cut-off; 3d, the release or exhaust; and, 4th, the compression or cushioning of the steam behind the piston.

CHAPTER VII.

PROPORTIONING PORTS AND SLIDE VALVES.

THE following is a simple, direct and comprehensive method of proportioning ports and slide valves for engines of any diameter of cylinder and variation of piston speed; also of getting the proper throw of valve, together with the time of opening and closing of exhaust.

In nearly all rules given, it is supposed that either the lap or travel of valve is first known. Now, the lap and travel of valve determine the port openings, and either of them *affects* the port openings. But they are, of *first importance*, instead of being treated as secondary.

The first thing to be considered, in designing an engine, is the amount of work it is to perform; after this, the size of cylinder, together with the piston speed, steam pressure, and point of cut-off.

Suppose it is decided that the cylinder should be 16″ diam. by 30″ stroke, the piston to travel 500 ft. per minute=100 revolutions, and the cut-off to take place at ¾ stroke. This engine would give 90 H. P. As the port openings must be in proportion to the size of cylinder, together with the speed of piston, they should next be found. We may get their area from Porter's rule—that steam, in entering a cylinder, should never be required to travel faster than 200 ft. per second.

In consideration that longer ports require larger valves and wider ports more travel, either of which creates more friction, it will be seen that port openings may be made too large as well as too small. We believe as good a rule as can be followed, for plain slide valve engines, is to suppose the piston to move at a uniform speed; the port to be at full opening all the time during admission, and velocity of steam 200 ft. per second.

The rule is:

Multiply the area of cylinder in inches by the speed of piston in feet per minute, and divide the product by 12000. This will give the area in square inches required for "live" steam,

$$\frac{500 \times 200}{12000} = 8.3 \text{ square inches.}$$

The exhaust steam should not have to travel faster than 110 ft. per minute. Make the steam port one inch wide, as shown in Fig. 22, and uncover it ¾″ by the "wing" of the valve. This is the full linear opening for "live" steam, while the exhaust has one inch linear opening. Divide the area 8.3 by ¾, which gives 11 inches as the length of port. The exhaust "throat" should be 2″ wide, for the reason that the slide valve covers it partly when at full stroke, as seen in Fig. 22. Make the bars ¾″, so that they will cast well. When the travel of valve is determined, the length, *e f*, of the face may be fixed. The valve should travel over the face at *f*, so as not to "shoulder." We have now found the full opening of port, and have decided upon the point of cut-off—¾. From these we can find the lap and travel of valve. Draw the vertical *y y* perpendicular to *a c*, cutting it in *e*. From *e* make *e b* equal to port opening. From 180° subtract one-half the arc traveled through by crank pin previous to cut-off. Divide the remainder by 2, and make the angle *e b c*= the quotient.

The arc traveled through by the crank pin previous to ¾ stroke is 120°, one-half of which is 60°. Make the angle *e b c*=

$$\frac{180°-60°}{2}=60.$$

Extend *b c* until it cuts *y y* in *c*. Bisect *b c* at *n*, and extend the perpendicular *n d* until it cuts *a c* in *d*. With *d* as a center, with a radius equal to *d b*, describe the circle *h c y*. From *c* draw *c d*—also *c g*—parallel to *a c*. *b e* is the port opening at full throw; *e d*, the lap; *b h*, the travel of valve; circle *h c y*, the path of eccentric; *c*, position of eccentric centers, when crank pin is at *h*. The angle *g d c*, is the angular advance of eccentric. The *c g*, the linear advance of the eccentric, *g i*, is to the travel of valve as that part of the piston stroke, after the closure and opening of exhaust, is to the whole piston stroke. The angle *c d b* is the one-half arc traveled through by tne crank pin previous to cut-off. Angle *d c y* measures the whole arc traveled through by crank pin previous to cut-off.

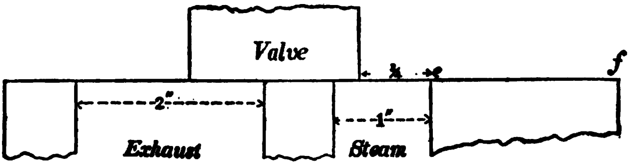


Fig. 22.

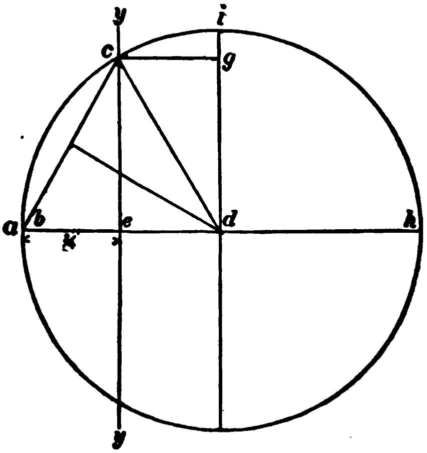


Fig. 23.

In the above there is supposed to be no lap on the exhaust side and no lead on the steam side. Neither is the angularity of the connecting rod taken into account. But the latter has nothing to do with the travel of valves or the proportions of valves and seats, but should be considered in the setting of valves.

By this simple process we have the following:

Cylinder 16'' x 30''
No. revolutions 100.
Cut-off, ¾ stroke.
Steam ports 1'' x 11''
Exhaust port 2'' x 11''
Length *ef* of face (Fig. 22) 2''
Bars ¾''

Throw of valve 3''
Lap outside ¾''
Full port opening ¾''
Angular advance of eccentric 30°
Linear advance ¾''
Exhaust closure 2.2''

CHAPTER VIII.

VALVE MOTION DIAGRAM.

THE practical man has no time for formulas when he can travel (to him) a much straighter road to what he wants. The plan is simple, and any one can understand why he does this or that; he knows he is right, and he has no conclusions to jump at.

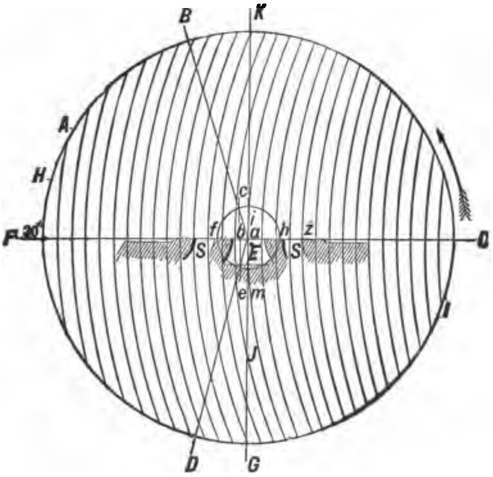


Fig. 24.

The outside circle is the crank-pin circle, the smaller one the eccentric's centre circle. The crank-pin circle is divided off into the inches of travel of piston, by a tram or pair of compasses set to represent the rod's length. *S S*, in Fig. 24, are the steam ports, *E* the exhaust port, and *X X* the bridges. Fig. 25 is a sectional eleva-

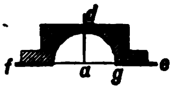


Fig. 25.

tion of the valve divided by the line *a d*. The valve drawing is cut out at the line *f e*, and placed on the line representing the valve-seat.

The laps and lead are laid off (from the pin *C*, the valve rod running directly from eccentric to valve) *a b* from centre of circles, *a*. The line *b c* is erected perpendicular to centre line *F C*, and through where it cuts eccentric circle at *c* a line is drawn, as *a B*. This line is the centre line of eccentric, and the angle *C a B*—the angle formed by eccentric and crank *c*. This angle is constant in whatever position the crank-pin may be.

Place the valve, Fig. 25, on the seat so that line *d a* coincides with line *c b*, and you have position of valve when crank is on centre. As the crank moves, in direction of arrow, from *C*, the valve moves from right to left, and back again, till right hand edge of valve comes to edge of port, *z*. The steam is now cut off, and, to find position of crank-pin, notice that the line *a d* on valve cuts the centre line, *F C*, at *b*, which extended down to small circle, cuts it at *e*. The line *a e D* is now evidently the centre line of eccentric. Taking the distance, *B C*, in a pair of compasses, and measuring back from *D*, we find the point *A*, which is the position of the crank-pin when steam is cut off. The figures on the large circle give the number of inches traveled when this occurs. The valve now continues to move from left to right till edge of exhaust cavity *g*, comes to edge of port *h*, when the exhaust commences. The centre line on valve *d a* now cuts line *F C* at *a;* this, extended down, cuts eccentric circle at *m*. A line, *a m j G*, now represents the centre line of eccentric; laying off the distance, *B C*, from *G*, we find the point *H* of crank-pin, when exhaust commences. The crank now travels on the lower half of its circle. The valve continues to travel from left to right, opening into the exhaust until the edge of exhaust cavity again comes to *h* (edge of steam port, the valve now traveling from right to left), the exhaust is closed and compression commences. This occurs, as before stated, when *g*, on valve, comes to *h*, on ports; the line *d a*, on valve, then cuts the line *F C* at *a*, and the line *a i K* represents centre line of eccentric. Lay off from *K* the distance *B C*, and we find *I*, the position of crank when compression commences. Similar points on the opposition stroke can be found by commencing with crank-pin at *F* instead of *C*, noting that centre line of eccentric is down instead of up. If the eccentric's motion is transferred to valve through a rocker shaft, the lap and lead must be laid off *toward* the crank-pin instead of from it.

These points, we assume, are understood by the reader, as it would require too much space to explain them all. The irregularity of the valve's motion (more expansion occuring on one end than the other) can be ascertained by this diagram. Unless the eccentric rod is *very* short, no notice may be taken of its irregularity due to radius (shown in line *b c*, being struck from length of eccentric rod), as it is so little that, generally, a straight line, as *b c*, will suffice for practical purposes.

We trust that this may be of use to some who have worried their brains with impracticable formulas and diagrams produced by men more scientific than practical, and which resemble puzzles.

MOTION CURVES.

The laying off of motion curves presents to the eye *all* of the movements of the valve at a glance. Fig. 26 shows the diagram after it is completed, ready to file away for future reference. Elevations of the ports are laid out to a scale and in length equal to the stroke of the piston, with the inches marked as shown. We start with the valve *V*, as shown at the commencement of the

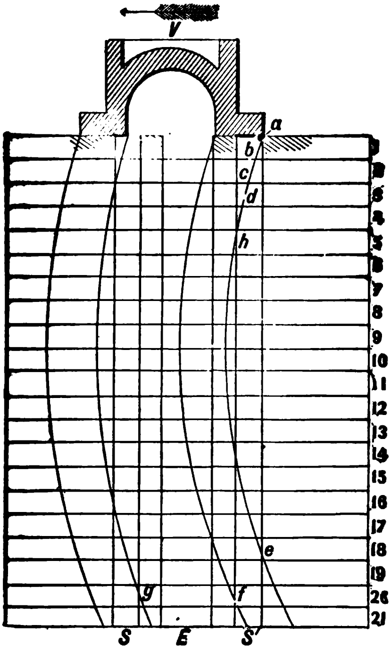


Fig. 26.

stroke, and, by means of Fig. 26, get the position of the valve for several positions of the piston, or for each inch, as shown at *a*, *b*, *c*, *d*. When all of the positions are obtained and jotted down on the port elevations, a curve that cuts them all is drawn, as are similar curves for the other edges of the valve. The opposite curves can also be laid down, and the movements of the valve for the opposite stroke obtained. In the diagram shown, we see that the port S′ was wide open at *h*, or 4½ inches of piston travel; that steam was cut off at *c*, 18 inches; exhaust opened at *f*, 19½ inches, and compression commenced at *g*, 19½ inches also, the valve being "line and line." Other points can be obtained by inspection, and the diagram forms valuable data for the builder and user.

TABLE BY WHICH TO FIND THE RELATIVE STATE OF PISTON AND EXHAUST AFTER EXPANSION.

Cover on the exhausting side of the valve in parts of the length of its stroke.	Steam cut off at 1/3 from the end of the stroke.		Steam cut off at 1/4 from the end of the stroke.		Steam cut off at 1/6 from the end of the stroke.		Steam cut off at 1/8 from the end of the stroke.	
	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke).
1-8th	.178	.033	.143	.019	.109	.008	.093	.004
1-16th	.130	.060	.100	.040	.171	.022	.058	.015
1-32d	.113	.073	.085	.051	.058	.033	.043	.023
0	.029	.092	.067	.067	.043	.043	.038	.033

Example from the Table.—Suppose an engine with a stroke of six feet, or 72 inches, and the steam cut off when the piston is one-third from the end of its stroke, the cover on the exhaust side of the valve being 1-32d of its stroke, the relative position will be the following:

In a line with 1-32d and under 1-3d is .113 and .073; therefore .113×72=8.136 inches the piston is from the end of the stroke, when the exhausting-port before the piston is shut, and .073×72=5.256 inches the piston is from the end of the stroke, when the exhausting-port behind it is open.

Table by which to ascertain the amount of lap necessary on the steam side of a slide-valve to cut the steam off at various fractional parts of the stroke.

To cut the steam off, after the piston has passed through

$\frac{1}{2}$ $\frac{7}{12}$ $\frac{2}{3}$ $\frac{3}{4}$ $\frac{5}{6}$ $\frac{7}{8}$ $\frac{11}{12}$

of its stroke. Multiply the given stroke of the valve by

.354 .323 .289 .250 .204 .177 .144

and the product is the lap of the valve in terms of the stroke.

Example.—Required the lap necessary to cut the steam off at the end of five-sixths of the stroke, the stroke of the valve being twelve inches, and without lead. .204×12=2.448 inches.

As lead is not taken into account because of different quantities being required to different applications of the steam-engine, subtract from the lap half the lead; the remainder is the lap required. Thus, suppose the lead equal .25÷2 =.125 and 2.448—.125=2.323 inches, the lap with one-fourth inch of lead as given.

CHAPTER IX.

HOW TO SET A SLIDE VALVE.

IT is first necessary to find the two centres for the cross-head. *A*, in Fig. 27, is the rim of the fly-wheel, and *B*, a piece of wood of a length to reach from the floor or foundation to about the height of the centre of the shaft.

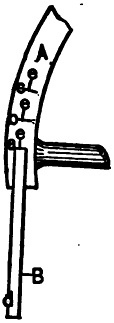


Fig. 27.

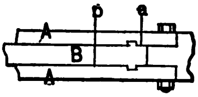


Fig. 28.

On the guides *A*, Fig. 28, draw a line *a*, at or a trifle beyond the travel point of the cross-head. Measure off a short distance (2, 3, 4 or 5 in.) on the guide from line *a*, and draw the line *b*. Place the cross-head as near on the centre as can be done with the eye, and turn the fly wheel, the crank-pin travelling up till the cross-head arrives flush with line *b*, as shown, *B* being the cross-head. Then with the stick resting on the floor or foundation, and against side of fly-wheel, make a mark *a*, Fig. 27, as shown. Now return the cross-head to the centre, revolving the fly-wheel in an opposite direction to that in which it was first moved, till the pin passes the centre and travels down, and the cross-head has again arrived at line *b* as shown in Fig. 28, the stick *B* remaining in the same position, another line *c*, may now be drawn flush with the end of the stick.

The centre between *a* and *c* may now be found and drawn, as *b*. Now, when the line *b* is brought flush with the end of the stick by turning the wheel, the cross-head will be on the centre. A similar operation for the other centre will also ascertain it.

The length of the eccentric rod is next in order, and should be lengthened or shortened as the case may be, till the valve opens one port about or nearly as far as the other.

While doing this the position of the eccentric makes no difference, and may, in fact, be fastened anywhere.

Having approximated the correct length of the rod, place the engine on the centre (either).

The direction in which the engine is intended to run being known, the position of the eccentric must be ascertained. If the engine has a rocker-shaft, the full part or belly of the eccentric will follow the crank-pin; if there is no rocker-shaft, the full part of the eccentric will lead the crank-pin, being in either case nearly at right angles with the pin. The engine being on the centre, as in Fig. 29, and having no rocker-shaft, we will suppose it is to run in the direction of the arrow. As the eccentric in this case leads the pin, it will be in the position of *A*.

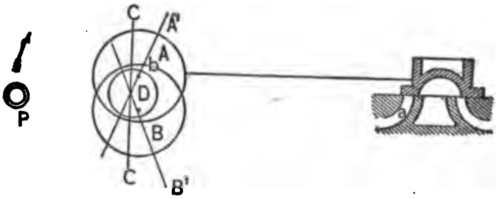


Fig. 29.

Now adjust it till the required lead appears at *a*, fasten it, and try the lead when the engine is on the other centre.

Whatever difference there is in the lead, lengthen or shorten the rod till the lead is equally divided on both ends, not moving the eccentric till this is accomplished, and when it is, shift the eccentric to the crank if there is too much lead, and from it, if there is not enough.

The centre of the eccentric *b*, will then be the lap and lead from the centre line *c c*.

If the engine was to run in the opposite direction, the eccentric would occupy the position of *B*. That the position *A* is the correct one, is evident from inspection, for imagining the pin *p* to commence and travel in the direction of the arrow, it is evident that the eccentric *A* will drive the valve from left to right, opening the port *a* as required.

If we connect the valve-stem to eccentric *B*, and suppose the pin to still travel in the direction of the arrow, the eccentric would drive the valve from right to left, and close the port *a*, and it is plain that *B* is not in the correct position for the direction of the arrow.

However, if the engine is to travel in the opposite direction, it at once becomes evident that *B* is in the correct position for that direction. In either case, the valve occupies the same position when the pin *p* is on the centre.

Locomotive engineers find in this fact an explanation for setting a slipped eccentric by throwing the reverse lever in the opposite direction to the slipped eccentric, marking the valve-stem, throwing the lever in the motion for the slipped eccentric, and shifting the eccentric till the mark on the valve-stem reappears in the same relative position as when it was made, as the valve occupies the same position when the engine is on the centre, whether the engine runs forward or backward.

By examining Fig. 30 it will be seen that the rocker-shaft makes all the difference in the world; in fact, the eccentric stands opposite to what it would without the rocker-shaft. When the engine runs in the direction of the arrow, the eccentric pulls the eccentric rod in the direction of the arrow under it, and drives the valve-stem in the direction of its arrow from the left to the right, as is required. The rocker-shaft just reverses the order of things, that is, it causes the eccentric to be the lap and lead *nearer* to the pin from a centre line *c c*, and also follow the pin, while without the rocker-shaft, the eccentric is as seen in Fig. 29, the lap and lead further from the pin, from the line *c c*, and it also *leads* the pin.

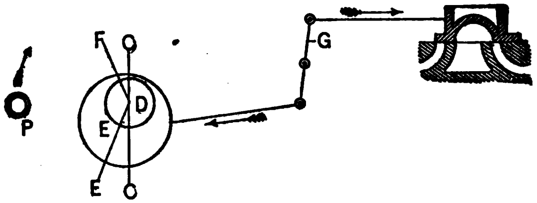


Fig. 30.

When the rocker-shaft is used, it should stand at right-angles with the valve-stem, when the valve is over the centre of the seat.

CHAPTER X.

ENGINE PROPORTION AND CONSTRUCTION.

WE shall endeavor to give our ideas of proper construction of the detail of modern horizontal engines, requisite in the machine for the faithful performance of its work as a motor for large manufacturing establishments. A stop of thirty minutes, caused by hot journal slides, or any part of imperfect design, entails a loss to the owners of from thirty to fifty dollars. It cannot be denied that this trouble is constantly occurring, and several cases have come to our notice within the past six months—one in particular, that of a five hundred horse-power double engine in constant trouble from hot slides, and will continue to be so until the surface on cross-head gibs is made equal to the labor, or according to any of the well-known rules on the subject of friction and lubrication. Then the water pans (we would suggest a more appropriate name—"bath-tub"), can be removed from the slides, and the owners of this costly machine can settle down to business without fear of from two to five stops a week. The slides and cross-head are not the only seat of trouble when treating of wearing surfaces is considered. The *hot* cross-head pin and crank pin are two common evils that our practical engineers deal with often, and seven out of every ten cases are due to bad proportion at these vital points of transmission, the trouble being that the builders make their cross-head pins not only too small in diameter, but entirely inadequate as to length. Now when the cross-head is *properly* guided, and provided with *complete* adjustment, this pin should be made one and one-half times its diameter in length. The pin itself should be made separate from the cross-head proper, then as soon as a flat or oval form from the oscillating movement becomes perceptible, the pin can be turned one-fourth the way round in cross-head, thus offering a new and perfect surface to receive the wear.

By making the pin long, we approach as near as possible (with a single end rod) the old forked connecting rod, still in use in Europe, and always free from heating.

The troublesome crank pins are, on the other hand, the *longest* ones. The great mistake is made in having too much distance from centre of strain to edge of support. Most builders make their crank pins from one and one-quarter to one and one-half times the diameter; and of course the long distance from crank face causes the pin to spring, and every sixteenth of an inch in length aggravates the disease. A crank pin running on a hundred horse-power engine, its length being twenty per cent *less* than the diameter; at the same time by the increased diameter the surface is ample for one hundred and fifty horse-power. *The pin was never hot*—but the trouble of all troubles comes when the heating selects the main crank shaft journals for its point of attack; this is liable to occur on any engine of size from the moment it makes the first revolution to the time it is ready for the second hand machinery dealer. Not always from badly proportioned journals and boxes, but from the fact that not one in ten of our engine builders provides any vertical adjustment for the crank shaft—it *will* wear *down* from its own weight, and that of the pulleys or fly wheels attached to it, and no man living ever saw both ends go down *alike.* Almost invariably, after three months running, the back end will be the lowest from one-eighth to three-eighths of an inch, most of this being due to the settling of the foundation. When the shaft leaves its horizontal, not only does the heating reach both main bearings, but of course the horizontal of the *crank pin* is also affected, and it at once falls into line and makes the agony complete, when twenty-five dollars extra expense on each box, in the way of a screw or wedge vertical adjustment, would place the shaft on its original level plane without loss of time or damage of any kind.

There are many good devices for taking up the wear of connecting-rod boxes, but the good kinds of adjustment are not the *cheap ones.* In the first place, the adjustments should be so arranged that the length between centre of pins will always remain the same. This assures the least possible amount of piston clearance in cylinder without danger of striking heads; then the taking up of wear should be accomplished by a gentle and moderate motion, and *not* by sledging on a key by main strength, thereby springing and otherwise injuring the brassess.

The moderate adjustment is not only safe on the brasses but safe in its working; it is never liable to drop out while the engine is in motion, and this is an item of vast importance with the new rapid motion machines. The *weight* of connecting-rod at centre and at crank end requires the most careful consideration, but is slighted too often in this respect. We have seen a rod on a one hundred and fifty horse power engine 20×48, making sixty-five revolutions per minute, springing by actual measurement $\frac{3}{16}$ of an inch at the centre at each downward throw of the crank, making the necks of this rod liable to fracture at no distant day. Not counting the injurious effect on the lower slide, twenty-five per cent. less weight in this case would have given ample strength.

THE SECURING OF PISTON-ROD TO CROSS-HEAD.

In the early days of the horizontal engine, the old style forked connecting-rod was always used, because it offered in the form of cross-head (which was wrought iron) a sure and easy fastening for the piston-rod.

The hub to receive the rod was made twice the diameter of the rod; the hole was bored to a taper of one-half inch per foot, and the length of hub was three times the diameter of rod. The rod was turned at the end to an accurate fit for the taper in cross-head, with thread on extreme end. It was drawn to and held firmly in place by a substantial nut, fitted well to the thread on the rod. But the forked connection for horizontal engines has since been supplanted by the single end rod, and consequently the nut on the end of the piston-rod has been abandoned, by reason of the inconvenience of screwing it up to place, and because its use added a few inches length to the cross-head. This change of front was, at the time it was made, a great mistake, and was so pronounced by many engineers. At all events, there has never been a better fastening used since, although there are several other methods in use at this date.

The screwing of a piston-rod into a *cast iron* cross-head, with a jam nut at the back of the cross-head hub, is about the most unsatisfactory device in use, but some of the most extensive manufacturers have constructed their cross-heads in this way for years. They fought against fate, however, in this matter of design, and we notice with pleasure their adoption of an old, but better plan. They now use either the double or single round end thin steel key. It is true that the key slot weakens the rod, but no more than the coarse V-threads they formerly used. By this latter plan the piston rod can be disconnected from the cross-head in one-tenth part of the time required in so doing with the screw arrangement, and there is no damage to the rod in either connecting or disconnecting when the key is used, while by the screw method the piston-rod thread is injured, more or less, every time it is removed from the cross-head. Then, in using the screw and check-nut fastening, the builders make their beds so narrow at the slides (the idea being compactness of design) that we have positive evidence that there are now in this country nearly two hundred engines, the piston-rods of which must be removed from the cross-head, and the cross-head removed from the slides and bed plate, *before the cross-head pin brasses, and connecting rod strap can be taken out for examination or adjustment.*

The above operation requires not less than one hour on a fifty-horse power engine, and the same time to replace the parts named. These facts lead us to the conclusion that the best way of making the piston-rod fast to the cross-head is by the old taper end design, with a wrought-iron nut, drawing the rod home against the taper, and a substantial shoulder. The nut can be cylindrical in form, as we now have several ways of screwing round nuts to a solid bearing in close quarters. Then the only change it makes in the cross-head is a few inches additional length. When the nut is in position, it is firmly fastened by a clip attached to the cross-head proper, and this precaution insures it against any liability of getting loose.

Piston rods should in all cases be made of machinery steel, well hammered. The allowance for turning from the rough bar need not be more than $\frac{3}{32}$ of an inch. Nothing is better adapted to the tensile and compressive strains of a piston-rod than the above-named metal. It is free to turn and finish, and polishes to a beautiful surface after a few weeks' running. The common, and perhaps the best way of securing the piston to the piston-rod is the shrinking process. It is certainly the cheapest; but one piece of machine work connected with this part must be done with great care—this is the *boring* of the piston head. The hole must be absolutely true and of equal diameter throughout its whole length, in order to have a uniform and perfect grip on the rod when cold after shrinking.

Two years ago the writer repaired a twenty inch cylinder engine, after a breakage from a loose piston. A few moments' examination disclosed the fact that the hole had been bored, by improper attention to the tool, $\frac{1}{60}$ of an inch tapering. The piston was seven inches deep, but had a bearing on the rod of only 1⅝ inches; the riveting on the end of the rod being too slight to offer any resistance to the loose piston. It would have cost no more to have made this a perfect job, in the first place, but it cost two hundred dollars to make it a perfect one after the break.

PISTON AND STUFFING BOX PACKING.

Various arrangements have been used to diminish wear and consequent leakage, partly arising from the adoption of high piston speeds. In Europe, where the piston rod of horizontal engines often passes through both cylinder heads, and is guided by a cross-head at each end, the solid phosphor-bronze piston has sometimes been used in cylinders up to about sixteen inches diameter. These pistons generally have a thickness of one-fourth their diameter, made quite light and partly sectional for adjustment. They have a series of small grooves, four or five in number, turned in their edge; the grooves are supposed to be filled by condensation, these acting as packing; but this piston has yet to be thoroughly tried.

The English use a solid cast iron piston with two or three small square steel rings sprung into the edge, but with these lubrication must be sure and perfect, or trouble will at once be at hand from serious and rapid cutting; but the piston receiving the most favor from English engineers to-day for cylinders up to thirty inches diameter, is the one with two cast-iron rings turned eccentrically so as to have a difference of about four to one between the thick and thin sides. The thin side is tongued and grooved together, and is not less than ⅜ inch thick. Both rings are reinforced on the *inside* by steel bands, and are fitted carefully to a central junk ring; they give no trouble and are very substantial.

Self-acting steam packing for pistons has not a very strong foothold in any of the old countries yet, but the good packings will probably take their places there in due time, and this turns us to the subject of self-acting steam packing in *this country.*

During the past twenty-five years, many patents have been granted, covering different modes of effecting a steam packing. Some of these have been practically introduced, and have shown good results when properly cared for; and when we say this, we do not wish our steam packing friends to think we are writing down their mechanical principle. Far be it from our thought to condemn any American invention, for the Americans are certainly the pioneers in this direction.

When the self-acting steam packing was first introduced, the steam pressure admitted to rings was allowed to be too great, and every cylinder in which it was used was worn to the form of an hour-glass, largest at the ends and small at the centre, owing, of course, to the high initial pressure at the ends, and the falling off of that pressure by expansion at the centre.

Cylinders with the common spring packing become largest at the centre, as at this point the piston reaches its highest velocity, due to crank motions, but not to an injurious degree. Makers of steam packing aim to neutralize the irregularity of wear, and have well nigh overcome the original difficulties by so arranging the steam openings to rings, that the initial pressure is much reduced and consequently retains nearly the perfect form of cylinder.

But the greatest difficulty met with in the matter of pistons is the *everlasting heavy casting*. As long as we American engineers refuse to guide the piston-rod for long stroke horizontal engines at both ends, we certainly ought to make our pistons as *light* as possible. Even of cast-iron they generally can be 25 per cent. lighter than the majority *are*, and if the steel casting, now plentifully made in this country, was to be substituted, a saving of 50 per cent. in weight could be made. We sincerely hope the day is not far distant when cast-iron *pistons* and *cross-heads* will bow themselves out for all time.

In examining and caring for the steam packing, the openings in face of piston, for the admission of steam, must be kept free from grease and dirt, otherwise the packing is rendered useless, and we have a leaky piston at once.

The natural twin of the piston-packing is the stuffing-box or piston-rod packing. In making a stuffing-box and gland *one thing is necessary in all cases, and that is the lining of box and gland with brass at every place where metal comes in contact with the piston-rod;* this is a sure guarantee against cutting, unless the packing is shamefully neglected.

The cavity for the elastic material used for packing should not be less than one and three-fourths times diameter of rod, and two and one-half times diameter of rod in length. The gland bearing outside of flange should be equal to one diameter of rod, but should have no more than one-half inch bearing in box; the remaining portion should be reduced in a conical form back to flange, making it at this point $\frac{1}{16}$ smaller than the end in the box. This will allow for any irregularity in screwing up the gland and will prevent any binding or undue pressure at this point. The methods of screwing up the gland are numerous, but the best is the cup-formed screw cap, because it does its work uniformly, and is easy to adjust while the engine is in motion. It is used but little on large engines, however, on account of its cost, although there are ways of making it at a cost very little more than the common flanged gland and stud.

THE CYLINDER, ITS JACKET, PORTS, ETC.

In speaking of this matter of proportion and construction, our readers will not set us down as a fault-finder.

Steam cylinders in this country have, up to a very short period, rarely been fitted with a live steam jacket. The subject of jackets being one of such wide difference of opinion among our engineers, the builders have, in nearly all cases, taken what they considered the safe side of a doubtful question, and have not used them.

We believe steam jacketing, when properly applied, to be a fuel-saving device; but it has so often been constructed without due attention to the collection of water from condensation, that it has no great achievements to boast of as yet.

A live steam jacket must be entirely free from water at all times, or it is not only useless in saving fuel, but will actually cause a loss at this point. In the first place, in arranging the steam and exhaust ports in a horizontal steam cylinder, the bottom line of passages should always be below the lower line of cylinder bore at least one inch, the points of removal of water from these ports and the cylinder ends easy of access, the openings of large area, and the automatic arrangement a simple, substantial, shifting valve made of brass entire.

The steam jacket relief must not be connected to the passage drainage at any point, and must consist of a large, perfectly reliable, self-acting trap, and every angle, side, end, or corner, entering into the design of cylinder and jacket, must have a downward course toward the trap. The outside case of the jacket must, of course, be covered with a good non-conductor, and under the above named conditions, the steam jacket is a benefit.

When the old speed of 350 feet travel per minute was the standard, one-sixteenth the area of piston for the steam port, and one-thirteenth for the exhaust, was considered the "proper thing," and a very good proportion it was. We know of some of the old engines, made by this standard as to ports, running to-day, not at 350 feet per minute, but at 520 feet travel, having been changed by request of owners and by mild consent of builders to increase power, and obtain the advantage in quick piston speed. What is the result? Initial pressure twenty per cent. below boiler pressure, and a uniform back pressure on the piston of from three to five pounds per square inch. We believe the most economical piston speed to be not less than 600 feet per minute. In some cases the pistons are speeded at 680 feet per minute, and we have found no difficulty in showing an initial cylinder pressure within three pounds of the boiler pressure, and no uniform back pressure.

We cannot see why so many of our engine builders still continue to guarantee certain duties as to fuel, at the same time never changing the ports in area to suit their much-increased piston speed. The laws regulating this matter of port area are just as rigid and unbending as those relating to the strength of materials, and must be observed in construction with the same care.

STEAM AND EXHAUST VALVES.

We do not think we shall here attempt to enumerate the various kinds of valves, induction and eduction, that have been used with positive and detachable motions during a quarter of a century.

The single-seated poppet-valve was used for many years in the old countries, but always under low pressure. This valve has been used in this country for a long period, especially on the Western and Southern river boat engines, and in these cases under very high pressures. Many ingenious methods for relieving the enormous pressure at the beginning of *lift*, or opening, have been used, one of which certainly deserves notice. This device consisted in the attaching of a small poppet-valve on, or very close to, main valve, opening into port or passage. A small toe or rock shaft *first* came in contact with the auxiliary valve stem a *little* in advance of main toe. Of course, the instant the small valve was raised, the pressure on large valve was rendered neutral, and it was picked up with perfect ease. These valves closed either by gravity or springs. The *double* poppet-valve was also used abroad in the early days of the steam engine, but its present form and manner of application and operation were left to our American engineers to perfect. They have modified this valve to such an extent that it is classed with the balanced valves; it can be made so near a balance that it leaves nothing to be desired in that direction.

The only trouble with a double poppet-valve outside of its cost in making is the matter of expansion. They *must* be ground to their seats when *hot*, and at the same temperature that they work in, or they *will* leak. We saw them used a number of years ago, in a way that obviated this trouble considerably, but not entirely. In this case the valve was brass, and was fitted into a brass cylinder that answered for the seat; the seat or cylinder was forced into a cast-iron chest, but was not fastened by screws; the cylinder had openings all around it for the free passage of the steam. Unless the steam was superheated to a degree that left little condensation at the chest, these valves were considered tight. Except the single port-flat slide, the *D* slide-valve is next on the list. It is used mostly for locomotives, and for stationary engines only where fuel is of little or no account. The *D* valve is in its *best* form when double-ported and partially balanced by its particular shape; the travel is then reduced about one half. We have yet to learn why it is not used more extensively by the locomotive builders. It is coming into general use on marine engines, and it certainly gets hard usage there, and no favors.

Then for effecting cut-off, there is the double slide-valve, one fitted to, and traveling on the back of the other, with independent eccentric, some of them balanced, others working under full pressure. This style of valve is arranged in a most ingenious and substantial manner, on the well-known quick speed Buckeye engines.

But the valves that we desired to speak of at length are those generally used in connection with disengaging valve motion. Of this class the Corliss is, with the exception of the poppets and common flat types, the oldest. The Corliss or segmental valve wears very well for two or three years, and at the end of that time must be replaced, *seats re-bored*, otherwise they will leak to a perceptible degree. A bad feature of this style is their being *single* ported; they require a long travel, which is very troublesome in anything like quick motion. The Corliss engine is rarely speeded over 70 revolutions, even the small sizes. A long traverse of a drop valve is also bad for steady regulating. The segmental valve cannot be called a bad one, but it is surely not one of the best. We all have our opinions on mechanical matters, which of course we are honestly entitled to, and we believe most of us have what may be termed a weakness for expressing our opinions, whether called on to do so or not. We will therefore beg pardon here for our opinion, that the best valve in use to-day on a detaching or automatic engine is what is known as the *sliding gridiron* valve with a number of *narrow* ports. The credit for the introduction and general application of this valve, both as a steam and exhaust valve, together with their chests, is due to Mr. William Wright, the well-known engine builder of Newburgh, N. Y. Mr. Wm. H. Hoffman made working drawings under this gentleman's supervision for valves of this kind twenty-five years ago. Some of these valves were made perfectly flat, others in the form of a V; the latter were fitted into the chest of old poppet-valve engines, the old valves, of course, being removed. They gave general satisfaction at that time, and have since been used with great success by the original inventor, Mr. Wright. At the date spoken of above, this gentleman built engines of 75 to 100 horse power, the greatest lift of steam-valve being *one-fourth of an inch.* These engines ran at about 60 revolutions per minute, but, as far as the valves were concerned, could have run at 100 revolutions without trouble, but the detail of transmission was not designed for high speed.

The piston valve (balanced) is in use both here and in Europe, but to a limited extent; its main fault is the requiring of nicety of adjustment, and it also must have a long movement. They are mostly used for steam pumps and hammers. But there are no valves that do not require care and watching. We are using high pressure, and in some cases super-heated steam in these times. Compound engines with 100 lbs. initial pressure are not expected to derive much benefit in the way of lubrication from natural condensation; they must depend on the *grease*, and that article must be at prompt call, and the movement of *all* valves must be brought to the lowest possible point.

VALVE MOTIONS FOR QUICK SPEEDS.

It is unfortunate that a limit in this direction is imposed, in most automatic engines, by the necessity of some form of independent detaching, cut-off valve gear, which admits of only a limited number of strokes per minute, regardless of the size or proportions of the engine.

The above presentation of this subject is certainly correct in every particular. It is very unfortunate that something in the way of a detachable valve motion to act with certainty and without unnecessary noise and wear, under rapid revolution, has not been perfected. We mean by rapid revolution 85 to 150 per minute. If there is a detachable gear answering the above requirements it has failed to come to general notice. As to the highest number of revolutions just named, we do not think it possible to introduce a perfect, or even ordinary working independent *drop* valve gear; but we do believe a detaching motion can be devised that will "pass muster" at 100 revolutions per minute. When this much-desired object is effected, however, the *lift* on steam valves will be materially reduced from the amount now generally given, viz.: from ½″ to 1¾″. The smaller figure is too much by at least one-third, for a two hundred horse engine, when the ports and valves are properly designed. Something of this kind would enable us to speed all engines with a stroke of from 30″ to 48″ as high as 100 revolutions per minute, if required, without fear of failure on the part of valve gear.

The engine of the future will be the one that satisfies the high or rapid motion requirements. The old 350 feet per minute and 60 revolutions are things of the past. The old idea about *non-compression* at the return finish was, by exhaustive and costly experiments, exploded in England and France ten years ago. Engineers then proved that *no* steam engine would work with its greatest economy without proper-sized steam and exhaust pipes and ports, quick *piston* speed and crank-shaft revolution, coupled with liberal compression.

It was fully established at the trials and tests just referred to, that it was a mechanical impossibility to reach the desired results in piston speed, crank revolution, and the attending economy of fuel, without a compression on the return to a point in pressure at least three-fourths of the *initial.* Otherwise the most important details of transmission soon perished, and *every* part of the engine suffered from undue strains.

We would like to ask the non-compressionist to consider for a moment that portion of the crank circle within 35° of its *finish* or central plane. Then consider the piston, piston-rod, cross-head, and connecting-rod, moving (at the speeds used in these days) with their *foot-pound* momentum, due to the velocity and weights, and we will then ask him if *nearly* all of that part of the crank's path indicated above, is not wasted energy, *without compression?*

If he does not utilize the vapor on hand, at this point, he will not only be chargeable with a loss at the cylinder, but will certainly have a terrible account standing against him in the matter of demoralized cross-head and crank-pins, to say nothing of the main journal. Compression has come to stay with us, and we must study to use it to the best advantage. The loss by extra valve leakage, in *short* stroke and *quick* revolution, is counteracted by the consequent saving in HEAT.

MAIN CRANK SHAFT AND DRIVING PULLEY.

In calculating the diameter of an engine crank shaft, it is necessary to consider: 1st, the strength to resist *torsional* strain, and 2d, its *stiffness*, or what is termed torsional elasticity.

It is well known that a shaft may be strong enough to stand all torsional strains that may reach it without twisting asunder, and at the same time be so *elastic* from its great length, as to render it entirely unfit to drive any machinery in which steadiness of motion is at all essential. It is therefore necessary to consider the strength of shafts and their stiffness independently, following, as they do, entirely different laws. Torsional strength varies as the cube of the diameter, and is wholly independent of the length, but torsional stiffness varies directly as the fourth power of the diameter, and inversely as the length. Or $d^4 \div L$.

Now rules proper for regular line shafting, whether placed in hangers for the ordinary transmission of power, or in larger sizes running in bearings bolted to solid foundations, are ***not*** proper for engine crank-shafts, owing, of course, to the irregularity of strains upon an engine shaft, due to the influence of crank motion and changing steam pressure.

Having the same labor to perform in foot pounds, the crank shaft must be sixty per cent. stronger than the line shaft when a single engine (one crank) is used; but with a double engine, or two cranks, one on each end of shaft at right angles, an addition to the torsional strength of twenty per cent. will answer, as far as that particular strain is concerned.

Our engine builders thirty years ago used cast-iron for crank shafts, as a forged shaft above six inches diameter at that time was worth from five to seven times as much as a cast shaft. The majority of the founders cast them solid, the result being that about one-fourth of the sectional area was *unsound*, and perfectly useless to resist any torsion. Others were wise enough to cast their shafts hollow, or with a core through them, removing fully one-fourth the sectional area, thereby adding strength to the remaining portion of shaft by the superior soundness given to it. The shafts were cast in a pit, on end, of course.

But the days of cast-iron shafts are passed; we get wrought-iron of the best quality in these days at a price of about double that of cast-iron;

they are much lighter, and are very stiff, from being well-hammered.

Of course the increased speed of engine crank shafts of the present day helps materially their transmitting power; for an 8 inch shaft transmitting one hundred horse power at sixty revolutions is one thing, and the *same shaft* carrying the *same power* at eighty-five revolutions is altogether another "individual."

But it is not a good plan to reduce too much in this direction, for the reason that the momentum of the various machinery *driven* causes a surging action which creates an excessive and irregular strain, and this must be provided for in fixing the shaft diameter.

In regard to driving pulleys, their size and manner of construction, there is much to be noticed, but we will not now consider the subject at length.

We must here enter a protest, and it is against the practice of *speeding-up* old driving pulleys twenty-five, and in many cases fifty per cent. above their original speed, for the purpose of increasing power of engine.

They are absolutely dangerous, and will destroy themselves, and perhaps life and property, by bursting—from actual centrifugal force—as soon as the laws relating to the *fatigue* of metals assert themselves, as they surely will.

Persons who take so much risk seem to forget that the centrifugal force of these old pulleys increases as the square of the velocity, so that in doubling the speed they have four times the amount of centrifugal force to contend with.

In many cases of these old pulleys the sectional area of metal is able to stand the increase, but there is always a hidden enemy that no man can safely estimate the power of. We speak of the strain brought upon a wheel in casting and cooling, being in halves generally, and consequently large castings. We all have seen apparently sound and beautiful pulleys, from ten to sixty inches diameter, burst asunder in the lathe, the very moment an ordinary cut was started, or by the simple matter of a fall of one or two inches. Even these pulleys, small as they are, are very dangerous when running (if they escape detection in the shop); but to what magnitude is this danger increased, when the diameters and weights are as great as they are in engine driving pulleys.

As we believe the best results from driving belts are obtained at a speed per minute of from 3,000 to 3,800 feet, we do not think it necessary to make any driving pulley for mills or general manufacturing over 14 feet diameter, and in most instances not over 12 feet (piston speed from 600 to 900 feet).

We believe also that all driving pulleys over 8 feet diameter, where *high speed* is used, should be made in *six* segments instead of *two*, and if they are well put together, can be run at any practicable speed without the slightest danger of rupture.

The extra cost is more than compensated by absolute safety.

No driving pulley should have a velocity of more than 75 feet per second, the point of rupture being with the ordinary casting a little more than 300 feet per second; this allows for a factor of safety 4, as far as vibrations and shocks from rapid speed are concerned, and according to rules above quoted, as relates to tensile strength or rupture, a factor guiding the construction of fly wheels and large pulleys little enough, all must admit; but there are old pulleys running to-day at the rate of 60 feet per second, and are consequently dangerous in the extreme.

DISENGAGING VALVE MOTIONS AND THEIR ADJUSTMENT.

When Mr. Corliss first adopted a regular standard of style or pattern for his horizontal engine cut-off gear, somewhere about the year 1853, he fixed on a plan superior in many respects to the standard now used by him.

Then there were no pieces of the detail of valve work attached in any way to the steam cylinder, save the four small connecting rods leading from trip motion to valve-stem arms.

All of the cut-off motion was fastened to the main bed-plate or frame, at a point nearly central with its length, and well away from the heating and expanding influences of the cylinder; and as the latch-block catches on barely more than one-eighth of an inch, it will be readily seen that all parts have very little leeway for disarrangement from the cause mentioned.

But the old plan has been discarded, and now every part connected with valve gear is attached to the cylinder, and is, of course, subject to all the changes by temperature common to the locality. Adjust this motion when engine is cold as perfectly as any valve setting can be, then start the engine, and, after an hour's running, take a diagram from *each* end, and they will show anything but proper setting; the conditions having been changed by the simple matter of expansion of metals. So it is absolutely necessary to make a final adjustment of this motion after the engine reaches its working temperature.

The exhaust valves of a Corliss engine seldom have lap and travel enough to give compression suited or required for *high* piston speed. These builders have their reasons for so doing, of course, which we certainly shall not question.

While the Wright engine (Newburgh), and the Brown engine (Fitchburg), both have their cut-off gear entirely assembled at the *lower* part of cylinder, they are not affected like the Corliss, as one has no wrist-plate, and the other, though a plate is used, has it so far removed from heat, and so short in connections, that no trouble is experienced on this point.

The exhaust valves of a Brown engine are operated by the best method ever introduced for this purpose. They are worked by a cam which throws the valve wide open just before the beginning of the return stroke, and there the valve remains until within a short distance of the finish of the return, when it closes as suddenly as it opened. To give the required amount of compression, the lingering of the valve is accomplished by a concentric spot on the cam. We have seen these engines running at very high speeds, and never saw over one pound uniform back pressure registered by the indicator.

There is one very important thing overlooked many times in adjusting and setting trip cut-off gear. This is the matter of ascertaining if the eccentric rod (whenever it is used) is the proper length. Simply because an eccentric rod grasps a wrist-plate instead of a valve-stem is no reason for failing to attend to this prime-mover. When a competent engineer prepares to set a plain slide valve, he first squares the motion and fixes the length of the rod exactly; he is then ready for the final touches. The leading motion for all detachable gear worked by an eccentric must have the same attention and correction of errors.

As for steam lead, we do not believe in it beyond $\frac{1}{32}$ of an inch. A valve motion too lazy in its movement to open a valve without wire-drawing is entirely unfit for use in these days, and we are happy to say that the kind just designated as lazy are very limited.

Our American automatic valve motions are all bordering a little on the complicated order, but they are so well made that this fault is lost sight of.

GOVERNORS AND THEIR SPEED.

The steam engine governor has, we believe, received its share of consideration by engineers, especially in this country. To say that our American mechanics have produced an absolutely perfect working instrument in the regulating line, would be putting it rather strongly; but when we claim that we make the best governors for steam engines of all kinds, we have the statement in its true and just form.

No English power carpet-weaving establishment ever had steady and economical speed until the American model and proportion of regulator was attached to their engines. The English plan generally has been to speed the governor *lower* than the engine. I would like to see them driving carpet looms on this principle. American engineers rarely speed engines in the manner just described, most of them driving the governor at the same speed as the crank shaft. The engines coming to our notice that regulate the best, are those furnished with governors speeded from 50 to 70 per cent. higher than the engine.

There are many patent *high speed* governors on the fly-ball, spring and water principle, all graded to run from three to six times the speed of crank shaft. But these, whether attached to their own *special* valves or to a detaching valve gear, fail to give the best results where the speed *must* be retained within *one* or *two* per cent. of the standard required.

An engine making 80 revolutions per minute, driving the class of machinery noted above, cannot vary more than one and one-half turns per minute without causing a damage to the fabric in weave room. There are engines performing this service to complete satisfaction, but they are not regulated by the extreme high speed governors. As a high speed governor, we believe the "Porter" to be as near perfection as it is possible to attain; it is certainly constructed on scientific principles, and is less subject to a trouble we shall speak of further on, than any other regulator of quick motion.

SINGLE AND DOUBLE HORIZONTAL ENGINES, AND THE TRANSMISSION OF POWER TO MAIN LINES AND COUNTER.

A wide diversity of opinion among experts and builders on the subject indicated above leads one to approach it with caution.

One of the constructing engineers in this country (Mr. Chas. T. Porter) would probably, should parties apply to him for a 1,000 H. P. engine, recommend, if high pressure was desired (without condensing), a single cylinder, 36in. diameter, and say 48 in. stroke, and about 110 revolutions of crank shaft per minute. If condensing, the cylinder would, we suppose, be from 30 in. to 32 in. diameter. He would in both cases cut-off at one-fourth the stroke. The driving pulleys would be 15 feet diameter, according to Mr. Porter's standard, though this proportion would not give as good results as smaller pulleys and wider belts, making belts double instead of treble. Thick belts do not have long life, according to our experience.

We will now draw comparisons between the single and double cylinder plan, not considering the compound engine system.

To develop an honest 1,000 H. P. with the two-cylinder plan (cranks at right angles) we would require each cylinder to be 26 in. diameter, with a stroke of 40 in. and 135 revolutions of crank shaft per minute (non-condensing). Beginning at the cylinder, we all know that the loss of heat in the one-cylinder plan would be much less than the double plan; likewise the friction, wear and tear, and valve leakage; while

110 revolutions per minute gives a motion steady enough for all purposes.

Looking, then, at the advantages attendant upon the use of the single cylinder, one can hardly see the use of at least 50 per cent. additional expense in selecting a double engine. There are, however, some details of engines and methods of operating connected with the double plan that are considered by some engineers to be superior to the single design. We suppose their reasons can be explained in a few words. In the first place, the crank shaft for the 36 in. × 48 in. single engine must be not less than 16 in. diameter, and about 21 feet long (without a middle bearing); the driving drums occupying 10 feet of the length, the weight of this shaft would of course be enormous—while the matter of lubrication of the large bearings would be an item requiring the closest attention. Now with the double-cylinder two crank plan, the main shaft to transmit 1,000 H. P., if made 12½ in. diameter (135 revolutions) would be ample, and would only weigh a little more than half as much as the 16 in. shaft: and lubrication would be comparatively easy, while the driving pulleys need to be but 9 feet diameter. Some engineers claim that it is always best to divide up the power of large engines, say above 250 H. P. (in pairs), in order to run only a portion of the work when desired; but with the exception of some special cases, we do not believe in arranging power with a view of adopting the system just mentioned. And now, as we have considered both plans to a somewhat limited extent, we will only say that the few advantages claimed for the double-cylinder plan, are more than outweighed by the real substantial ones of the single-engine system, with a centre bearing on main shaft; while, with our improved and perfect method of lubrication, all fears of serious trouble from heating and cutting may be dismissed.

We will refer to a few methods of transmitting power from engine to line shaft or shafts. The practice of gearing an engine to and speeding up a jack shaft, should be avoided in all cases; likewise narrow thick belts; likewise belts running more than 4,500 feet per minute, as 3,800 feet will give the best results in the long run. Not long ago we saw a 550 H. P. engine attached to a single line shaft (full load on and a little more) —if we remember rightly, there was "music in the air," which was continued until the power was led to two line shafts.

Further, 1,000 H. P. should be transmitted to work through at least three line shafts on counters, and four would be better. If three are used, the driving belts should each be double and 40 in. wide (speed 3,800 feet per minute is the best for economy), the counter and line shafts at the point of receiving power supported in self-oiling bearings on a firm truss gallows-frame constructed of channel and beam iron; bearings universal, and three and one-half times the diameter in length. The counter or line shaft pulleys should be split and keyed to shaft, the keys one and one-half times their depth in width. A pulley 40 in. wide should not be crowned more than ¼ of an inch; if belts will not run with that allowance, something is wrong with the setting up. And we would suggest the making of driving and counter or line shaft pulleys with the rims mortised to receive well seasoned white pine segments, to be turned off true and smooth and bolted after being driven into the rim. This method would add 25 per cent. to driving power or traction of pulley, while the rims would be much lighter, an important item as to line shaft at all events. In conclusion, we will add that the use of two or more driving belts to take the power from large engines, is an absolute safeguard against accident, as all of the belts would not break at once.

STUD BOLTS AND THE PRACTICE OF TAPPING CAST IRON.

We are aware that in assembling and erecting the parts of a steam engine, we are compelled to use some *stud bolts*, and must necessarily screw them into cast iron, and we are just as well informed of the fact that more than half of the stud bolts generally used for the purpose under notice, could be dispensed with, at a decided advantage as far as labor and strength are concerned.

Nearly every builder uses stud bolts all the way around on cylinder heads, and yet most of them could use the T head bolt, except perhaps in one or two narrow spots. The holes for these bolts can be drilled close to the body or wall of cylinder so that the pressure will be brought near to the *joint*, and the cylinder heads need *not* be increased in diameter. With the T head bolt, if one is broken how easy it is to replace it if duplicates are kept on hand. But some one will say: Stud bolts are all right if you only look out for them when screwing up. Yes, they are all right, *until* some man comes along with plenty of main strength and *too much wrench*, and quietly twists them off, and that is about the time they are *all wrong*.

Then there are the main crank shaft bearings, seven out of every ten of which you will find fitted with stud bolts, a locality where it is much more convenient and vastly cheaper to use the T head bolt let in flush at the sides of the boxes. And again, the bolting on of the various stands and brackets is common in design. In this matter, there is not an engine built to-day but might be so designed as to use plain square headed bolts, the heads in many cases let in their depth in casting to prevent turning when screwing up. But how many engines do we find fitted as here suggested? Few, we assure you.

If it is the *correct* and *proper* thing to screw 3 inch cap bolts into *cast iron* frames and blocks, and 1 inch and 1½ inch stud bolts into castings in various places, what is the use of having *head* bolts with nuts? Why not, we say, if the practice is correct, make *cast iron* nuts for our large bolts, instead of going to the expense of forging them with such care as we generally exercise? No, we know that no *sane* man would think for a moment of making a cast iron nut for a 3 in. bolt, and yet he will screw that same 3 in. bolt into a *casting* where it was easy for him to use a T or square head, and this is not an overdrawn picture at all.

But it is to be hoped that this universal practice of *tapping cast iron* will be discontinued in the near future. Let the builders start with the determination that they *will* discard *stud bolts* in every case *possible*, and the result much needed will be reached.

BOILER PROPORTION.

Settings for Boilers.

The settings for shell tubular boilers must, for the sake of economy, be largely constructed of brick: but it is, we believe, the general opinion of our American engineers, that the ordinary setting, as now constructed, has too many bricks in contact with the boilers.

In the first place, a boiler should in no way depend on the brick work for its support; but the lugs riveted on each side of the shell should rest on upright posts made of two pieces of channel iron, thimbled and riveted together, thus forming a light, strong, durable column, besides providing a complete universal adjustment for the expansion of the boiler, as the tops of the columns are free to sway in any direction; the bases being held in a cast iron foundation plate.

The bolts for holding fronts in position should not be anchored to brick work, but should be secured to the first set of channel iron columns supporting boiler.

Every fire-tube shell boiler should have two walls on each side of boiler, leaving a space of from 3″ to 5″ between boiler and inside walls, and 3″ space between the two walls.

By this means a way is open for the expansion and bulging of inside wall by heat, without affecting the outside wall in the least degree, while the non-conducting air-space is a great source of economy. Between the boiler and inside walls there should be a light flange of cast iron resting by wrought iron straps on the channel iron columns, and closing the space between boiler and walls at this point, thus preventing the brick work from coming in contact with the boiler.

This is one of the most important items to be considered in setting any boiler.

The buckstaves for holding side walls in position should not be of cast iron, but of channel iron, with the flat web placed next to brick work, the lower ends bedded in concrete and the upper ends tied together (opposite sides), by bolts one inch in diameter passing over the return arch on top of the boiler.

The return arch should be made either of brick or blocks of asbestos, or some other non-conducting material, resting on T iron rafters, bent to the form of boilers; the return space under arch to be not less than 16″ high in the clear for a 66″ boiler, to admit of thorough cleaning and inspection.

The main bridge wall, directly back of grate bars, should be wide, for strength to resist the blows from fire-tools, and hollow, to admit air in sufficient quantity to ignite the escaping gases. Just back of this bridge-wall, there should be a small combustion-chamber, about one-half the capacity of the furnace where hard coal is used, and a little more than the full capacity of furnace where soft coal is the fuel. The object of this chamber, of course, is to allow the oxygen to freely mix with and ignite the unconsumed gases drawn over from the furnace. The remaining passage from combustion-chamber to return connection at the rear of boiler should be quite close to shell of boiler, on a 66″ shell. This distance should be no more than 12″ and no less than 8″. The spaces and combustion-chambers under shell fire-tube boiler must not be too large, else there will surely be a loss from excessive radiation.

There should be cleaning-doors on one side of boiler where room will permit, but always at the rear of boiler, and these should be large enough to clean the tubes as well as the fire connection and spaces under the shell.

The cast iron fronts must not be too thin, on account of their liability to warp, but they can be well ribbed, and thus, in a measure, prevent this trouble. Where they are bolted together, the holes should be an eighth of an inch larger than the bolts, to allow for the expansion of castings. All fronts should be made in *four* pieces, planed or chipped together. When this method is observed, they will *never* crack at the *doors* or sides from excessive heat.

The setting should be done in a substantial manner, using hard brick with sharp lime mortar. The furnace and bridge walls should be laid with No. 1 fire brick, every fifth course a header. It will make a good job and assist in keeping it clean, if the space back of bridge wall is smooth paved; and the outside appearance may be further improved by putting a stone coping on outside walls. If the boiler is to connect with chimney, a breeching will make the connection across the front. If an iron smoke stack is used, do not let the weight of the stack rest upon boiler front, but put on a petticoat where it passes through the roof of boiler house, and let part of its weight rest upon the roof.

FEED AND BLOW-OFF PIPES AND VALVES.

An 80 horse-power economical engine will require an evaporation by the boiler of 1,800 lbs. of water per hour, for the cylinder alone; and an additional amount of at least 200 lbs. per hour for loss by radiation, etc.

The perfect feeding apparatus should distribute this 2,000 lbs. of water uniformly, as to time and quantity, through the space of sixty minutes, and into the body of water in the boiler at a neutral point or plane of temperature.

The feed pipe should not enter the boiler near the bottom for several reasons.

The first is the bad effect this method has on the *fire-sheets* of shell, by sudden contraction, as the water reaches the sheets where the fire is fiercest, at a temperature 150° below that of the water already in the boiler, assuming the feed water to be heated by heater to 180°. Another great objection to entering the boiler (with feed) at the bottom, is the matter of sediment, or, in many cases, quantities of mud, that rapidly collect at this point, and almost wholly obstruct the feed passage; anything in the water liable to produce a deteriorating chemical action, will of course attack the fire-sheets at once, and fitting and grooving is the result.

Then, again, the feed water should not be delivered too near the water level, as all foreign substances in the water will, at this locality, promote priming, together with the natural disturbing elements thus near the surface where the steam takes its final leave of the water; besides its tendency to affect the temperature is still more, so far as the steam is concerned, than at the bottom of shell.

Steam should leave the boiler through a perforated pipe.

Now the feed water should enter the boiler at the central plane of the temperature of the body of water, through a *fine grade* perforated pipe screwed into a seat-plate bolted to front head of boiler, the pipe passing through the entire length of boiler; and this pipe should be so arranged and fitted, that it can be easily removed from the boiler, when steam is off, and cleaned. It should be supported in two places within the shell, and should be turned and bright on the outside throughout its whole length.

Now the holes in this pipe should be in the lower side, so that the entering currents of water will unite and travel downward with the circulating cool currents already in motion, as they help to induce rapid circulation. The holes in the pipe should not be larger than ⅛ of an inch in diameter in three rows.

When these conditions are complied with we have the minimum loss by feeding. Let any disbeliever try this once; it will not cost him much, and will surely pay him. The blow-off pipes should consist of two, connected of course, but independent in action. The lower one should be at the extreme bottom of shell, the upper one within three inches of the mean surface of water. This pipe should be 3″ in diameter, perforated with ⅜ holes, and nippled down on the outside of head to 2″ diameter (for a 66″ shell); the pipe must extend the whole length of boiler. This constitutes a reliable surface blow.

The inside perforated feed pipe should have a connection, outside, with the 2 inch blow-off pipe, in order to clean it now and then by blowing through it.

Every check valve should be so constructed that it can be raised from its seat by hand from without, and its position shifted; by this arrangement its correctness of working can also be tested. All blow-off valves must be of the screw-lift system, either globe or gate pattern. We are now comparing gate with globe valves. The great trouble with globe valves, besides their cramped and angular lines of passage, is their want of proper area, compared with the pipe that matches them. A 4″ globe valve, as you find it in market, has a clear space of 9½ square inches, and a 4″ pipe has an area of 12½ square inches. All boiler mountings clear of the shell outside should be brass, to resist corrosion and to present a neat and cleanly appearance, and the drippings and discharges of water gauges, and gauge cocks, should be carried entirely clear of the shell, head, and fronts, to a suitable drain.

GAUGE COCKS, WATER GAUGES, SAFETY VALVES, AND STOP VALVES.

The improvements in boiler mountings have nearly kept pace with those of boiler design and construction. Allowing this to be a fact, it must be admitted that the changes have been very slow in producing beneficial results.

Among the great variety of gauge cocks, where is there one that is not in some degree troublesome? Beginning on the inside of the appendage under notice, it is perfectly safe to say, that nine-tenths of the styles in use to-day are liable at any time to clog and be useless upon very slight provocation. As the water falls to the line of opening with the average style gauge cock, all that is necessary to render this important fixture entirely unfit for service is a few slight particles floating on the surface, or water level, and any attempt to free the cocks under pressure can be properly termed a "red-hot" job. They can only be thoroughly cleaned by removal from the boiler after steam is off. Now, a gauge cock can be made that will have a half or three-quarter-inch hole all the way through it, to within, say, $\frac{3}{8}''$ of the front end, the remaining portion having a hole for discharge, say, $\frac{1}{8}$ or $\frac{1}{10}$ inch in diameter. Then, just back of the small opening at front end, there should be a two-way three-ported cock, so arranged that when the small aperture at any time becomes closed by gradual, or even sudden filling up, the $\frac{1}{2}''$ or $\frac{3}{4}''$ two-way cock can be closed to front end, and the same movement will open the discharge at once to the blow-off drain pipe. The obstruction can then be removed from the front end without danger of burning the operator, if it is not blown out through the drain.

A gauge cock should never enter the boiler head or shell, but should be screwed into a liberal-sized water cylinder. For an 80 H. P. boiler, this cylinder should be $3\frac{1}{2}''$ diameter inside, and connected to the boiler head or shell by two $1\frac{1}{2}''$ pipes. By this means, all agitation from circulating currents and priming can, in a great measure, be avoided.

The bottom end of water cylinder must have a 1¼″ blow-pipe and valve fitted to it, and discharging into the blow-off drain pipe. The cylinder should be a steel or brass casting, about ⅜″ thick, and, for a 66″ boiler, 24″ long. The glass water-gauge will also be attached to it, as well as one of the steam gauges. *Every boiler* should have *two* steam gauges. The water-cylinder has been used by some builders for years, but most of them are very much too small, as well as the pipe connecting them with the boiler.

Regarding safety valves, we have been surprised to find the variety of sizes placed on boilers of like capacity by different builders. On a 60″ shell boiler, the valves vary from 2½″ to 4½″ diameter. Now, there are $4\frac{9}{10}$ square inches in a 2½″ safety valve, and $15\frac{9}{10}$ square inches in a 4½″ valve; so, according to these areas, some one is, "all at sea" in the matter of safety valves.

In the first place, every boiler should have *two* safety valves; one, a lever and weight pattern, the other, a spring, or pop valve. The opening in *each*, when raised to its limit of lift (which is always very small), should be large enough to prevent the blowing-off pressure being exceeded more than ten per cent. The size or area of safety valves depends, of course, on the amount of water evaporation with a certain quantity of fuel in a given time; and it is a well-known fact, that no safety valve raises over seven per cent. of its diameter—or, in other words, a 4″ safety valve will never lift more than ¼ inch, and 1/12 inch is the usual amount for this size valve with the ordinary working pressures.

Therefore a 2½″ valve would have to raise more than double the distance of the 4″ valve to relieve the pressure in the same time, and in case of sudden closing of stop valves, or an undue pressure from any cause, the 2½″ valve would be liable to *fail* to relieve the pressure in season to prevent the straining of boiler and perhaps its explosion, if defective.

A 66″ shell and 14 feet boiler, 54 tubes, 80 H. P. should have two 4½″ safety valves, and neither will ever lift over ⅛ inch. They will be *absolutely* safe, if examined daily, as *every valve* about a boiler should be. The weight lever must never be longer than the regular working standard pressure requires. There can be no overloading then, unless extra weights are used, and any man who will resort to this vice should be introduced to the paymaster without delay.

The majority of safety valves have their seats fitted on a bevel. The *spherical* form of seat is much better and relieves more readily, while its face can always be automatically shifted a little on the seat. The lever pins should work on knife edges, thus offering the minimum resistance to rapid opening.

There should also be two stop valves on a

boiler, one a globe valve, the other a balanced lever valve. The locomotive throttle style is good. The lever valve should be used only in case of an emergency, such as the racing of engine caused by derangement of regulator, breaking of main belts, bursting of steam pipes, etc.

The handle or lever should be so arranged that it could be operated either in engine or fire rooms. It requires too much time to close a *screw* valve, in case of accident.

All stop valves should be 25 per cent. larger than the pipes to which they are attached, in order to provide a free passage through the necessary bends, etc.

Fig. 31 shows a working drawing of a steam engine with description of cylinder, jacket and valves for a 100 H. P. engine designed by Wm. H. Hoffman, M. E., of Passaic, N. J., one of the most talented engineers of this country, and one to whom the writer and all other American engineers are indebted for much valuable information in the design and construction of steam engines and boilers.

The cylinder is completely jacketed with live steam taken direct from the boiler through a 1¼″ pipe. The jacket casing is of boiler plate $\frac{3}{16}$″ thick, riveted on to the cylinder, leaving a steam space of ¾ inch all the way around the cylinder. The surfaces on cylinder and jacket are perfectly smooth, thus offering no obstruction to drainage. *No water can lodge in this jacket.*

The cylinder heads also act as jackets, live steam being admitted to each, and each is drained at the bottom.

All water from condensation is trapped into a small tank, and is from thence forced by a small pump, working continuously with the engine, into the boiler.

The main steam valves are connected by a right and left screw for perfect adjustment. They are balanced up to two-thirds of the chest pressure, by rings fitted to the chest bonnet, the lower rings being made of cast iron, while the upper ones are a series of thin plate rings, each $\frac{3}{32}$ inch thick, convexed and concaved faces, placed together for elasticity. The rings are brought to the proper bearing by four set screws. The space inside the ring forming the balancing surface is $\frac{1}{16}$ inch deep, and is connected to the upper end of the small double-acting drain pump, which maintains a vacuum under the rings and by which any leakage can be detected at once.

The main valve is worked by an eccentric, which transmits its motion to the valve stem by a peculiar lever and wrist-plate combination, giving a very rapid movement to main valve at the point of opening and past the half stroke, when it begins to reduce its speed, and at full port pauses and moves toward the close very gradually, until the half travel is passed again, when it closes, as it opened, with great rapidity. By this means a port of twenty square inches (which is the size at the cylinder bore) is just as efficient as 26 square inches would be with a simple eccentric motion. The ports are each $\frac{13}{16}$ inch wide in each valve, outside lap $\frac{3}{4}$ inch, inside $\frac{3}{32}$ inch. Travel of valve $3\frac{5}{16}$ inches.

The cut-off valves are also worked by an eccentric and wrist-plate motion. The travel of these valves can be changed at will, but for usual work they will travel four inches. They are operated direct by a centrifugal governor fixed on the main crank shaft. By the lever and wrist-plate combination used in this case, the cut-off valves have a rapid close by the cut-off eccentric, independent of the extra hastening of the closing by the governor. Both valves can be worked by hand lever (the main and cut-off), and the cut-off valves can be adjusted on the seat of the main valve to any desired point, by a simple device entirely outside of the chest.

SETTING UP STEAM ENGINES.

In setting up stationary engines, the first requisite is a template of the anchor bolt holes. This template is made of one-inch stuff, about six inches wide, and is merely a frame large enough to take in the holes in bottom flange of bed plate, as shown in Fig. 32.

The engine may be tilted up a few inches

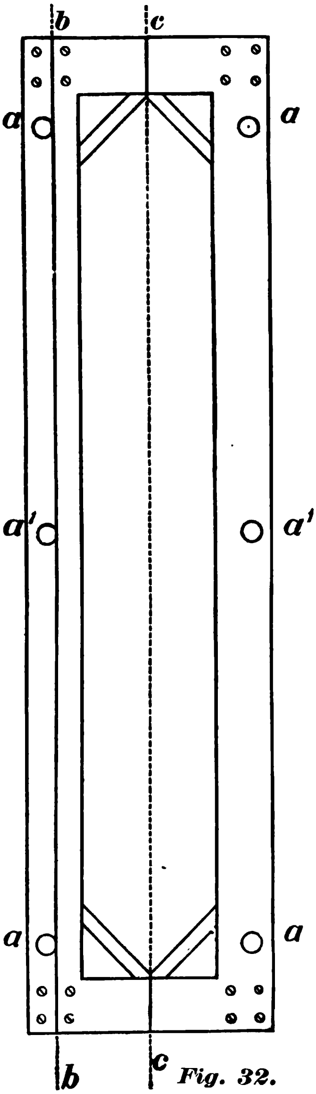


Fig. 32.

on one side and supported by short braces from the top flange. This will allow of the template being slipped under the bed plate and the flange holes marked on it through the flange holes. While in this position, if the side or edge of the bottom flange is planed true with the centre line of engine, a line *b*, *b*, Fig. 32, may be scribed on the template true with this edge, to square the anchor bolts by an existing line of shafting in the building, as the case may be. If the top flange of bed plate is planed and the bottom one not, the line *b*, *b*, may be squared down from top flange. If neither the top nor the bottom flange is planed on the edge, a line may be put up through the cylinder centrally, extending the entire length of bed plate, which should be level or nearly so, and a line dropped from such end of this centre line across the ends of bed plate and the bottom flange, marked fair with the line where it crosses it. This transfers the centre line of engine to bottom flange, which in turn can be transferred to the template. The marks on lower flange had better be made while the engine is in the shop on the "floor" being put up. The lines so transferred are shown in Fig. 32 at *c*, *c*. The holding down or anchor bolts, which should have large heads, may then be slipped through the washers or anchor plate. A heavy anchor plate of cast iron, long enough to take in two opposite bolts, as a^1 a^1, is better than single washers. The bolts so prepared may then be placed upright in the foundation hole, the template resting on the nuts, which should be run down a short distance, as shown in Fig. 33, *d*. being the template. The template and bolts

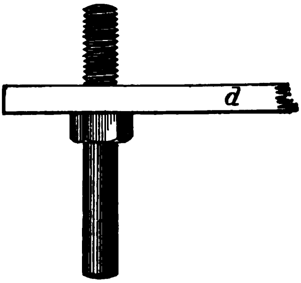


Fig. 33.

should then be squared by the lines *b*, *b*, or *c*, *c*, (Fig. 32,) to the shafting, if up, or the building, and braced in this position by several light braces tacked to it and surrounding objects. The foundation can then be laid on the anchor plates enclosing the bolts till the required height is reached. The top should be composed of large cut stones and carefully leveled. The engine, after the foundation has "set" sufficiently, may be placed on it, the bolts bringing it in line. It should be carefully leveled by iron wedges. A "runner," or small embankment of clay, should then be built as high as the top of the lower flange, both inside of bed plate and out, and extending entirely around it about a quarter of an inch from it. This "runner" may be bent out at several places into the form of a capital letter *U*, open side to the flange, to facilitate the pouring of the melted sulphur or lead, which should be run up considerably higher than the bottom of the flanges, so that the hydrostatic pressure may keep it solid between the flange and stone. Cement, either Portland or Louisville, may be used in place of the lead or sulphur by "pointing" the crevice between bed plate and stone on the outside only with a thick mortar of the cement, and pouring a thinner mixture of the same on the inside. This may be worked beneath the flange with a trowel. After it is set the anchor nuts may be tightened, and the surplus sulphur, lead, or cement chipped off flush with the bed plate.

The writer has seen several engines set without the template, by building up the foundation solid without the bolts. The engine should then be leveled on the foundation and squared. Holes may then be drilled through the flange holes into the foundation, about 12 or 14 inches wider at the bottom than top. The bolts, which should be slightly tapered, being smaller at the top, may then be placed in the holes and sulphur or lead used as in the first plan. An 80 H. P. engine set by the latter plan six years ago is as solid today as can be wished. Sulphur will generally give the best results. When the latter plan is adopted, the foundation throughout should be laid up in cement. In setting up an engine it should be remembered that a foundation too heavy (if such a thing is possible) is preferable to one too light. The part below the earth should be laid of stone well below the frost line. The part above may be of brick topped with one or two cut stones. The pillow block foundation may be built the same as the engine foundation, and, being lighter, anchor bolts clear to the bottom had better be used.

STEEL VS. IRON PLATES.

In the year 1847, Krupp, of Essen, made his first steel twelve-pounder field piece, and all of Europe's military engineers laughed, and said the "Dutchman" had better try glass next: but in 1879, the same Krupp made a seventeen hundred-pounder breech-loading rifled siege gun, the weight of which is but little more than one-half that of the famous one-hundred-ton gun, made of wrought-iron, built in England, while the projectiles and powder charges are about the same weight—and Europe's military engineers don't laugh as much as they did. The trouble in 1847 was, the great German engineer had not perfected his theory, but he did in 1879. Consequence; no more wrought-iron guns.

The history of the steel plate is about the same as that of the gun, as far as its history has progressed, and we are only about two-thirds the way toward the completion of its troublesome journey. The first steel boiler plates were used about eighteen or twenty years ago, and were, of course, too hard and failed; in the next trial this difficulty was, in a measure, modified and overcome, but the sheets were sadly wanting in homogeneity, and could not be depended on, as two plates were rarely alike. The plate, at this time, was made quite thin, in fact, too thin; not for tensile strength, but it was inadequate as to mass or body, to resist the intense heat to which it must be subjected, for all carbonized metals must be liberally constituted in this respect or they will surely fail, from the fact that the main stay will be burned out of them, if they are put to hard work. Then, again, there is a decided limit, at present, in the matter of thickness, for it is very difficult to obtain steel plates, say ⅝″ thick, that can be called perfect, a majority of them being bad in spots at the centre of thickness.

But steel plate making is slowly and *surely* improving. The famous mills of Pennsylvania are marching steadily on to the solution of the problem; they have been but comparatively a short period at this great task, but they have accomplished wonders. Therefore, in considering the subject of Boiler Proportion and Construction, we shall introduce steel in construction at all points where it has already been faithfully tried. Of course, the cost of manufacture of a steam boiler using steel in most of its details is materially increased: first, in the raw plate; second, in working in the smith's shop; finally, in the machine preparation. Being carbonized, the utmost care must be observed at the flange-fire in working the steel plate, the fire must reach the metal at all points exposed alike, or the plate will be rendered unfit for service, and it must not be subjected to sudden shocks or blows in bending. Steady pressure and close attention must take the place of rough usage.

In machine working, the caution must be equal to that required at the fire. *No* steel plate should be punched; drill all holes, otherwise the main object is sure to be defeated. A punched steel plate is little, if any, stronger than a drilled wrought-iron one of the same thickness, and a punched hole, any way, is always enlarged after the boiler is put under pressure. In riveting, the hydraulic system should be universally adopted for steel plates, on account of its freedom from concussion, and in places that exclude the use of the machine and that call for hand work, the heavy blows should be administered at the insertion of the rivet, the hammer strokes gradually growing lighter as the rivet cools; excessive hammering must be avoided, else the rivets are to a certain degree crystallized.

TUBES VS. FLUES—EVAPORATIVE POWER, ETC.

According to the recorded experience of a large number of American engineers, the contest between tubes and flues, as generally arranged in boilers where the heat passes *through* them, has thus far been decidedly in favor of the tubes when properly proportioned and located.

The construction of the tubular boiler, however, has often been inferior to that of the flue design.

The original idea of the tubular boiler was to fill a shell as nearly *half* full of *very* small tubes (say 1½″ diameter) as possible, and from actual compulsion, locomotive engineers are even to-day building their boilers after this style, knowing, of course, that their main source of power lies in the artificial draught by the exhaust. Otherwise the small tubes would not only be inefficient in the amount of surface, but they would utterly fail to take the volume of hot gases through them, being simply wanting in area for natural draught.

The old-fashioned stationary tubular boilers soon showed their weak spots; *incrustation* was the death of this method of arranging tubes.

As the water in boilers becomes concentrated the order of disposition of the different salts therein is as follows: 1st, carbonate of lime; 2d, sulphate of lime; 3d, salts of iron as basis or oxides; 4th, silica; 5th, common salt. Carbonate of lime considered as chalk, marble or limestone is held in solution in fresh water by an excess of carbonic acid, or it really is a bicarbonate; the acid is driven off by the heat, and the carbonate is precipitated. Sulphate of lime and carbonate of magnesia are also precipitated in all kinds of water, in fact, on becoming soluble, the lime and other salts remain in a partial state of suspension in the water, but gradually deposit themselves according to the density of the water.

Now, a *flue* boiler with the lower flues close together, and near the bottom of shell, is just as objectionable as far as the precipitation and deposit is concerned, as the old small tubular style; while the real danger from burning out, and explosion, is far greater than with the small tube system, as neither can be perfectly cleaned.

As to the evaporative power of each system when compared, take for example, a two-flue boiler 48″ diameter, 24 feet long, 15″ flues; now, take an old style tubular, same diameter, half filled with 2″ tubes, 14 ft. long; now start with a sixty-day trial, both boilers *clean.* The flue boiler at the first week's trial would evaporate 6½ lbs. of water with a pound of ordinary steaming coal, while the tubular would evaporate 8½ lbs. more water with one pound, same fuel. At the *last* week's trial the flue boiler will have fallen off to 5¾ lbs., but the tubular will fall as low as 7 lbs., both boilers being under the same conditions as to setting, draught, firing and feed water—the loss in both cases being due to natural deposit of a poor conductor of heat on the most available heating surfaces, and when this deposit is removed it involves the laying off of these boilers for four or five days, and the cleaning is called finished when it can really only be imperfectly done. Now this all goes to prove that *no* boiler should be termed economical that cannot be thoroughly cleaned wherever the fire reaches it, both *inside* and *outside.*

THE LOCATING OF TUBES AND FLUES.

This natural effect can only be overcome or neutralized by either chemical, galvanic, or electric combinations, or actions.

But there are methods of locating flues and tubes, or even fire-boxes and stays, in boilers, that cause more trouble, damage and expense than anything chargeable to elementary causes. The badly designed and proportioned boiler, clean or unclean, will seriously interfere with the generating of power, independent of all other conditions.

When tubes, or flues, are placed so near to each other, or to the shell of boiler, that they actually prevent circulation, and the separation of the steam and its free passage from the water, we have a mechanical difficulty before us insurmountable.

We do not wish to make an unreasonable statement, but we really believe that two-thirds of the boilers in use to-day have their tubes and flues too near together—too many of them for the diameter of shell; tubes too small in diameter for their length, where natural draught is depended on; while in fire box, or internally fired boilers, the water spaces are not only too narrow, but it is utterly impossible to clean them.

We will look at the arrangement of tubes first. The diameter of horizontal tubular boiler shells, now-a-days, varies from 30″ to 96″, and the lengths run from 10 feet to 24 feet. The tubes used are from 2″ to 5″ outside diameter. The fire, of course, passes through the tubes.

A boiler with a 66″ diameter shell is generally 16 feet long, and contains about sixty-six (66) 3½″ tubes—too many tubes by *twelve*, and too long by 2 feet, for the best results. The diameter of tubes for the shell is all right.

It is not proper, by reason of natural causes governing the transit of heated volumes of gases, etc., to make tubes proportionate in length to their area. For example, a 5″ tube should ***not*** be double the length of a 3½″ tube simply because it is double the area of the small tube.

Experiment and observation have taught us that, in doubling the area of a tube, the advance in length can only be 50 per cent., with economy in view, so that a boiler with a shell 96″ and tubes 5″ diameter should be 21 feet long—no more, no less—for the best steaming duty. In all boilers, tube or flue style, the tubes or flues must *never* be nearer the bottom of shell than 10″, while the side rows of tubes should clear side of shell at least 4″ (we are now speaking of boilers over 48″ diameter). They cannot be cleaned unless this rule is observed.

In designing a 66″ x 14 ft. tubular boiler 3½″ tubes, we would place the lower course of tubes 13″ from the bottom of shell, and the side tubes 4½″ from shell. Tubes 1⅜″ apart measuring on a *vertical* line, and 1⅝″ apart measuring across the horizontal. The top course is entirely above the centre of boiler; no steam drum, but a liberal-sized inside dry pipe, short, and placed near the front end of boiler; boiler inclines toward the front 3″. A boiler constructed after the above design, properly set and mounted, will generate steam for an 80 H. P. engine without the slightest signs of priming, and there will be no more than *fifty-four* (54) tubes, and fifty-two (52) will give the power noted above; and the water can never be driven away from the bottom and sides. The tubes should never be staggered.

Some of the steam drums placed on steam boilers are certainly ludicrous in the extreme. They are not only useless in preventing priming, but they act as condensers, render the boiler weak from the mode of attaching them, and are ungainly and a nuisance in handling. As generally constructed, they are about as useful as an ordinary quart measure would be in their place. A boiler that has enough water and steam space in its shell requires no steam drum, and it never will prime, to a noticeable degree, unless sadly overworked.

Priming is, we believe, caused, to a large extent, by a reduction of pressure and increased ebullition under that point of the boiler whence the steam is drawn. By this means, water is carried into the very top of steam drums, and we have seen cases where the inside of a drum would be plastered with mud two inches thick, and not a particle of it on the tubes, or inside of shell, except, perhaps, a little in the immediate locality of the drum. A steam drum cannot be protected or jacketed so as to prevent the reduction of temperature and pressure below that of the boiler shell proper.

MAN-HOLES, STAYS, HAND-HOLES, SIDE SEAMS, CAULKING, ETC.

The use of cast iron man- and hand-hole frame, and plates should be discontinued. Our rolling mill engineers can do almost anything in the line of irregular forms at this date, and there is no excuse for using a metal entirely unfit for the details mentioned. About one-fifth of the boiler explosions almost daily occurring, can be traced to the use of cast iron in construction. We have seen man-hole frames on boilers carrying 80 lbs. pressure, that if actually tested, with a view to rupture, would break at 130 lbs. distributed in the same manner as the steam load. Let all builders adopt wrought iron for the parts here under consideration, discountenance this dangerous practice, and the record of boiler explosions will surely be smaller. When wrought iron is used for frames and plates, two man-holes can be placed above the tubes, in shell, with as much safety as one, each one say 11″ x 15″. The hand-hole in front head *under* tubes should never be less than 7″ x 10″, on a 66″ shell boiler we would make it 9″ x 12″, the common size hand-hole is of very little use in cleaning, never being larger than 5″ x 7″, the majority 4″ x 6″.

The *head* stays above the tubes should be continuous in design, with eye sockets and pins at the heads, the rods or bars separated in their length at the centre, and coupled and drawn together with taper keys, wired in position, to prevent displacement. The stays can then be taken out and examined when cleaning, an item of as much importance as any other part of boiler inspection. The bolts and clamps, or guards, for holding plates in position, must be large, the threads of bolts of the half V style to resist hard usage, while the clamps should be *faced* on feet and hubs. All boilers should be double riveted on the horizontal seams. Even with this precaution in construction, they are not as strong at the point of riveting as in the solid portion of sheet.

As for caulking, Connery's concave system will be of as much service in boiler making, in the matter of saving material and preventing accident, as any invention introduced within the last twenty years. Hundreds of good boilers have been ruined in caulking, and many a boiler maker has repaired his own work, knowing, after examination, that *his caulkers* were responsible for the damage. Machine tools must be used in preparing sheets, such as straightening, bending, flanging, drilling and scarfing, or planing, and in *assembling*, riveting. The old sharp-edged caulking tool was always wrong, the same as hand riveting is in principle, and, in the hands of an inexperienced man, the most dangerous of all devices used. It will pass away, however, and good results will certainly follow its disappearance.

Mr. James W. Connery, of the Baldwin Locomotive Works, deserves honorable consideration from the mechanics of this country, for his scientific and simple method of dispensing with a mischievous article of *destruction* as well as construction, and we hope he may be pecuniarily benefited in the same proportion that he has assisted the mechanical world in boiler construction.

STEEL PLATES—PUNCHING VS. DRILLING.

Passing from the consideration of the foregoing points, to another important branch of the subject of boiler construction, we will speak of steel plates and their advantages.

The best steel plates, though quite mild in quality, show entirely different results in working from wrought iron.

Any attempt to do even ordinary flanging by *hand*, at more than two heats, is liable to be met with failure, and one heat shows better results on the plate when cold.

The portion of plate to be turned must be wholly exposed to the fire, and that at as high a temperature as safety will admit; while good wrought iron plates can be heated a portion at once in several heats if desired, only a moderate fire being necessary.

In handling steel plates, if proper machinery is used, it is much better to bend or turn at *one* heat; the plate is then perfectly safe from straining or fracture, besides the chances of burning are very much reduced. The tensile strength of steel plate for flanging should not exceed 78,000 lbs. per square inch of section, and working on this basis, a saving of from 20 to 25 per cent. can be made in the weight of a boiler, where steel is used instead of wrought iron. Another decided advantage in using steel plates, is that the rivet holes do not *stretch* under heavy pressure as much as iron, a very important point, as it is under any circumstances almost impossible to fill a rivet hole perfectly, on account of the shrinking of rivets while cooling. A great deal is said about the grip the sheets get, by the shrinkage of the rivets in the direction of their length while cooling. This is very true, and the same law that shrinks them in that direction, operates in reducing their *diameters* to such an extent, that seven out of ten rivets do not fill their holes when cold; and when you add the stretching of an iron *punched* sheet to this difficulty, you have a riveted joint, ripe for shearing under sudden and undue pressures or shocks.

In places where the fire is in direct contact with the boilers, we do not believe in making the sheets much less in thickness than that now generally adopted when *iron* is used in construction, and if corrosion is the same with both metals (steel and iron), which is doubtful, the thicknesses must be about the same in each case; but there is very good reason to believe that the small amount of carbon in steel will prevent in a great measure the wasting of the sheets by bad feed water. This is now a disputed point, however, which will in due time be practically decided without any serious clashing of engineering opinions.

In regard to the effect of punching on steel plates, it can be stated with safety that steel plates of suitable grades are far more liable to be injured by punching than wrought iron. Builders of steel boilers, who punch their plates, generally ream the holes to a standard size for rivets, or at least they *should* do so, and if they would anneal the plates after punching, they would further restore the strength, in a great measure destroyed by punching.

But we cannot see the use of punching steel plates, and then adding another expense in two extra operations to get the plates back to anything near their original strength. We believe that all steel plates should come from the rolling mill annealed to their proper working condition, as the mills are fitted with annealing furnaces or ovens constructed on scientific plans, and most of our boiler shops are lacking in this particular at least. Then, in preparation, they should be drilled by a machine similar to one designed by Mr. W. H. Hoffman, the eminent engineer. This machine carries twelve drills, vertically arranged, of course, in a carriage which is automatically fed downward with an adjustable feed, drilling twelve holes at one operation, the drills being returned after the down stroke by a quick automatic return motion. The drills can be spaced or separated from 1¾ to 2¾ inches from center to center. The tables for holding plates are also moved automatically and by hand when desired.

Eight years ago Mr. Hoffman proved, to the satisfaction of several of our expert boiler makers, the correctness of the principle of drilling both steel and wrought iron plates, by designing and constructing an automatic universal machine for drilling and tapping locomotive fire boxes, *screwing in* the stay-bolts, and then cutting them off. We believe the machine is now at the celebrated Brooks Locomotive Works. It is thoroughly portable, being on small trucks, and is adjustable in every direction, the drill traveling-head having three motions, and is fitted with a universal ball chuck for holding drills, stay-bolts, and cutting-off tool. The machine is furnished with universal clamps for holding it to a great variety of work outside of boiler construction; in fact, it is a boiler and machine shop tool, power being transmitted to it by a small rope.

When the improved machine tools in boiler making once get a firm foot-hold, the wonder will be that they were not adopted years ago, for they certainly have been talked about and dreamed over for more than a dozen years. When they are finally accepted as standard tools, the boiler will then be on the same plane of perfection in the matter of construction as the engine, and its life will be materially lengthened, and the boiler builder will be as proud of the finish and accuracy of his work as our modern engine builder of to-day.

When wrought iron plates are used and punched, the drilling machine should have come into play, reamers being inserted in the place of drills, and the holes finished to a standard size. Without this precaution the boiler is an imperfect one.

Steam riveting can be applied to wrought iron sheets with safety, but the hydraulic process should always be used on steel plates, and as little hand riveting along with it as possible.

RIVETING AND STAYING.

It has been determined by many experiments that the use of steel rivets increases the cost of construction, by reason principally of the loss by overheating and burning, while in hammering they often chill so rapidly that the heads are easily knocked off; this trouble showing itself even when mild steel is used. A steel rivet should only be used where the system of machine riveting is on the hydraulic plan.

Machine processes in all cases close the joints and fill the holes better than hand riveting, as the whole body of the rivet is reached by the pressure of the machine, but we believe that in every kind of machine riveting, pressure rings surrounding the rivet on both sides of the plates should be used to assist in bringing the joint home and to prevent the sheets from being separated by the squeezing of the rivet while very hot, in a transverse direction under or between the sheets, though in hydraulic riveting the pressure is so gradual that the partial cooling of the rivet prevents it from "bulging" and separating the joints.

In drilling or reaming plates it is easy to slightly round the corners of holes on both sides of the sheets, thereby reducing the tendency to shear all of twenty per cent., and punched holes should have the corners reamed in like manner.

There is one point in the construction of a boiler that often receives very slight attention, viz.: the forming or forging, and fastening of crow feet or stay lugs to shell and heads. It is not a rare thing to find crow feet forged of flat bar iron strong enough to resist with perfect safety a working strain of 40,000 lbs. fastened to the shell of a boiler with two ⅝″ rivets and these receiving the stress at an angle of 45°.

We think that a neat and strong crow foot can be dropped or rolled with two, three and even four webs to receive rivets; these can be heated and bent to any form desired. This method would certainly establish a uniform standard as to strength, and would materially reduce the cost of manufacture in the shop, besides the advantage in having room for extra rivets.

The stays should be of the best material hammered from the merchant bar. Holes in the ends rimmed to size. The stay pins should be turned and drilled to receive annealed wire guard pins. Where threads are used the parting of rod on which thread is cut should be enlarged, so that the bottom of thread will have the same sectional area as the body of stay rod. Continuous head stays should have steel expansion collars to allow for expansion of tubes. Collars must be of steel. Every stay rod should be so arranged that it can be removed from the boiler and examined and tested, and wherever pitting and wasting are discovered the stay should be condemned, and a perfect one inserted.

Now some boiler builder will say this is all very fine with your hammered stays, turned pins, rimmed holes, etc., but who will pay for it? But nevertheless, my friend, this is just what boiler-making is coming to, and the purchaser will pay for the extra work and reduce his rate of insurance; and with boilers well designed with a view to perfect cleaning and inspection, perhaps he will not insure at all, as he will naturally feel as proud of the boiler as of the engine, and nearly every owner of a good engine will insist on its being regularly inspected and cleaned.

THICKNESS OF SHELLS, ETC.

It is seldom that we see horizontal boilers of late construction less than 36″ diameter. The 1½″ and 2″ tubes have been supplanted by the 2½″, 3″ and 3½″ sizes.

A shell for a 36″ boiler, made of iron, should be ¼″ thick, and it is best *not* to reduce this thickness, if made of steel, for reasons already mentioned. The horizontal seams should be double-riveted *always;* rivets ½″ diam., pitched 1⅝″, or 1¾″, from centre to centre. Close or fine pitch of rivets insures *tight joints*, and does not require as much calking. Of course the strength is also increased, not only by the additional number of rivets, but by increased adhesion of the plates.

The shell of the above-stated size of iron will sustain a working pressure of 100 lbs. per square inch, and of steel 125 lbs. per square inch. Boiler heads, in both cases, 3/8″ thick. Shells 48″ diameter, of iron, should be 5/16″ thick, but 1/4″ will answer, if made of steel; rivets pitched 1 7/8″ and 9/16″ diameter; double-riveted boiler heads 7/16″ in both cases; working pressure, in both cases, 90 lbs. per square inch.

For shells 66″ diameter of iron the thickness should be 7/16″, and, if of steel, 3/8″ thick; rivets pitched 2″ and 5/8″ diameter; boiler heads of iron 1/2″ thick, of steel, 7/16 thick; working pressure, in both cases, 85 lbs. per square inch. Shells 96″ diameter should be 11/16″ thick, of iron, and 9/16 of steel; rivets pitched 2 3/16″ and 3/4″ diameter, double riveted; boiler-heads of iron 3/4″thick; of steel, 5/8″ thick. Working pressure, 80 lbs. per square inch.

Where flues are used instead of tubes, a 12″ iron flue should be 3/8″ thick. If of steel, 5/16″ should be the thickness for 100 lbs. working pressure in both cases. For a 36″ *flue*, cornish or internal fire styles, the thickness of iron should be 5/8″, and of steel 9/16″, provided the length is no more than 22 feet. Flues over 24″ in diameter should never be made longer than eight diameters, and those from 15″ up to 24″ in diameter no more than twenty diameters in length, and seventeen to one is nearer the absolute safety point, considering the flues, of course, as exposed to *external* pressure, the working standard of the 36″ flue being 80 lbs. per square inch.

Large flues *must* be thick, as they cannot be properly stayed if *thin*, without making the inside of boiler a wilderness of stay rods and bolts, thereby cutting off the availability of thorough cleaning. Increasing the thickness, therefore, will dispense with a great deal of staying.

Boiler explosions are occurring daily. Two and three in a single day is not a rare notice in the daily papers, and among these are many *new* boilers. What is the cause of these terrible accidents? We hear of a tug boat being blown to atoms, all on board perishing. The same day a locomotive sends destruction through the air at a railway station without an instant's warning. The boat and the locomotive were both in charge of reliable and competent men. Again we ask, What is the cause? Is it not faulty or careless construction and proportion? We think most of the trouble can be charged to these causes—not altogether from a lack of knowledge of design and strength of material, but really from a fixed idea of false economy, which seems to have a firm hold on the purchasers. We have an abiding faith in the judgment of our American mechanics and constructors, and believe that, if they can only get a fair price for their work, they will give, in return, a safe and durable boiler.

SPACING RIVETS, WIDTH OF LAPS, ETC.

On a previous page where we mentioned the pitching of rivet holes, we referred mainly to *steel* sheets, and wholly to *drilled holes.* Where punching is the system used in construction, the rivet (pitch remaining the same) should be increased fifteen per cent. for steel sheets, and twenty per cent. for iron sheets.

Arranging the rivet holes for double riveting, and adopting the triangular form, or zig-zag pitch, the sides of the triangle should never be equal, but the base on lower row must be that of two right-angled triangles, forming, of course, a triangle whose perpendicular is one-half the whole base. Therefore in the case of $\frac{9}{16}''$ steel plates (96″ shell) using ¾ rivets (double riveted joint) the pitch according to standard before mentioned, measured on the hypothenuse, will be $2\frac{3}{16}''$ while the whole pitch at the base will be 3″. The *perpendicular* distance between rows, of course, will be 1½″, and the lap or total width of joint can be now fixed, as the distance from rivet hole to edge of sheet will be one and one-quarter times the diameter of rivet; so the joint or lap will be $4\frac{3}{16}''$ wide. Now a wide joint adds strength to a boiler, and requires but very moderate caulking when closed up with a hydraulic riveter.

The great object to be reached when putting plates together is to have the shearing strength of a double row of rivets equal to the remaining section in the sheets. The tendency to rupture at the sheet is most at the section under the rivet, or from the hole to the edge of sheet. Many boiler makers fix this distance less than the diameter of rivet. This method is wrong, and is liable to split or rupture at this point, as the rivet is acting like a *blunt wedge* to force the metal apart. The excessive driving of steel drifts to "line" the holes has a very weakening effect at the point between the hole and edge of sheet, and is one of the worst and "wickedest" tools of a boiler maker's "kit." This, and the old sharp cornered caulking tool, should be abolished, as there are more ways than one to make rivet holes "line."

In fastening stays for crown sheets and large internal flues or flat surfaces, the riveting of holding lugs or crow feet must be well distributed. The strains must not be *concentrated* on the shell of *any* boiler. Instead of lugs and crow feet, bars of T-iron should be used, rivited with tolerably fine pitch to the roof of shell, so that the whole arch will receive the strain, and the shell consequently be strengthened by the T-iron.

Now while a boiler is under a static force, constant pressure, it is all right; but when that force is turned into a dynamic current, and that current is intermittent, with pulsations ranging in number fifty to four hundred and fifty per minute, then the conditions are vastly different. As we note these pulsations indicated on the steam gauge by a fall, *twice* in the revolution, of from one to five pounds, we pass the poor old boiler without even a glance while *it* is going through the same range of pulsations as the steam gauge pointer; the only difference being that while it does no harm to the gauge, it on the other hand has a very injurious effect on the shell. Take the case of the locomotive that exploded a few years ago on the Morris and Essex Railroad at Hoboken. Allowing this engine to have run five hours each day, with an average revolution of drivers of 150 per minute, this would subject the shell to 180,000 pulsations per day, and if there was bad bracing the laws relating to the fatigue of metals must certainly have asserted their rights. We believe the sheet ruptured, in the explosion mentioned here, was near the crown sheet, a little to one side of centre, and received the crow-feet lugs in a straight horizontal line, just below where was a double riveted side seam very stiff in itself. The action then was to buckle the sheet at each pulsation as the cylinders took steam, between the side seam and crow feet; the stays and double seam below being the two fulcrums. The parting of sheet in *two* places, one very near the crow feet and one near the double seam, establishes the correctness of this theory.

HEATING SURFACE.

The evaporative power of all boilers must be measured, of course, by the efficiency of their surface exposed to the fire in transferring the heat from without the plates to the water within them. Heat reaches the plates by radiation and contact. The contact comes from the solid fuel, the radiation from the flames and gases.

Water has a weak conducting power, and, if heat is applied to its upper surface, it does very little toward warming the water beneath, and, for this reason, the heat should be applied to the bottom of any vessel to produce the most effect as to evaporation. In case the enclosure is surrounded by water, the heat being applied from within, the greatest amount of evaporation takes place at the top of casing. Where the water is within and the fire without, the best surface is the curved style of crown sheet of a locomotive boiler. The next best surface is the horizontal plate, though seldom used on account of its weakness in design. The next in efficiency is the large cylindrical tubular boiler shell (the under side), where the heat first strikes it. The top inside surface of tubes and flues has also high evaporative power. The bottom of a tube or flue evaporates very little water, as the water will not leave the under side of the tube unless driven away by forced circulation. With a 3½" tube there is six times more water evaporated directly *over* the tube than at a point directly *under* it, while the sides of the tube will evaporate a little more than one-third of the quantity evaporated at the top of tube. We are now speaking of boilers that pass the heat *through* the tubes. Then, again, the heat does not travel in true parallel lines through the tubes, but has *wave curves*, with a decided tendency to hug the upper side of tubes.

Where the water is in the tubes, and the fire and heat around them, the order of things is reversed, and the bottom side of tubes in the most powerful generator, the top side yielding a very small evaporative duty.

Sectional boilers, with water tubes inclined to induce circulation, give the greatest evaporative power, but the life of them is much shorter than the shell and fire tube system. A properly proportioned and constructed *fire* tube boiler will evaporate from 9¾ to 10 lbs. of water per 1 lb. of coal, and a good sectional *water* tube boiler will evaporate between 11 and 12 lbs. of water per one pound (1 lb.) of coal, both systems working under the same conditions as to grate surface, firing, feeding, etc.

The shell fire tube boiler will last, with ordinary care and fair feed water, twenty-five years, and very little repairing will be needed during that time; but a sectional boiler of same capacity will fail, with the best of care and feed water, inside of twelve years, and frequent repairs will be necessary at the joints and connections.

We speak of the sectional water tube boiler as it is generally designed and constructed. Further on we shall speak of the sectional system at length, though we will here say that we believe that no water tube should be less than 5″ in diameter; else, when arranged on the sectional plan, the water will be driven from the lower courses of tubes and the metal burned to an injurious degree. When a *shell fire* tube boiler is well designed and proportioned, nine (9) square feet of heating surface will produce a horse power, with an allowance of one (1) square foot of grate surface for every thirty-two (32) square feet of heating surface.

FIXING THE NUMBER OF TUBES.

We have already stated that most of the boilers now in operation have too many tubes for the best economy.

This conclusion was reached by Mr. Hoffman by actual experiment. Of fifteen boilers experimented on, all using well proportioned heaters, and above the average steaming coal, only two gave an evaporation of nine pounds of water per pound of fuel, and only *one* $9\frac{4}{10}$ths pounds. Four touched $8\frac{1}{2}$ pounds, two $8\frac{1}{3}$, and the remaining six ranged from 8 down to $7\frac{8}{10}$, which was the lowest.

The highest evaporation was reached with a 66″ boiler, containing sixty-six $3\frac{1}{2}$″ tubes, fourteen feet long; eight of the lower courses of tubes, and *four* of the side courses being purposely closed at each end. Before the change was made, this boiler gave an evaporation of $8\frac{9}{10}$ pounds to one of the fuel, and after the change the evaporation was $9\frac{4}{10}$ pounds.

Before the change the tube area was $\frac{22}{100}$ of the grate surface, and after the change $\frac{17}{100}$ was the area compared with grate.

The heating surface before the change was 39 to one, compared with grate, and after the change, $35\frac{1}{4}$ to one. The temperature at the flüe leading to chimney (chimney 120 feet high) before the change was 462°; by the change this was raised to 540°, and the damper was shifted from three-fourths open to half closed. The velocity of the hot air was, before the change, about 122 feet; after the change, this was increased to 136 feet per second. This goes to show that before the change there was too much tube area for the grate (although the grate surface was a little too small). The extra amount of tube surface offering too much resistance, by friction caused the currents of hot air to be sluggish, and, of course, the temperature fell below the economical point, and the heat did not impinge on the sides of the tubes with a sufficient degree of intensity. We think it is generally conceded that the gases cannot leave the boiler at a point below 520°, without impairing the draught power of the chimney.

In the case under notice, this power measured in inches of water 1.112; this was after the number of tubes had been reduced. The cold air entered at about 70°.

Therefore, we believe that the tube area should be about ⅐ of the grate area, and sub-divided as little as possible, consistent with good circulation, and making *no* tube more than 48 diameters long.

The boiler giving the lowest evaporation had a 42-inch shell, and contained seventy 2½ V tubes, *staggered* tubes being fourteen feet long. This boiler was primed continually, and none of the usual remedies seemed to affect it. The tubes were ¾ of an inch apart, and, under ordinary firing, a vibration of the tubes could be noticed, caused, of course, by the forced currents of water through cramped passages.

Other boilers with sixty-four 3½ inch tubes 18 feet long, gave only an evaporation of 8⅓ pounds; and of those with 3″ tubes, 16 feet long (60″ shell), 74 tubes gave 8 pounds, and one of the same size with tubes, staggered or zig-zag, $7\frac{7}{10}$ pounds.

The fuel in all cases was hard coal of various brands, of quite uniform quality, *Buck Mountain Lehigh* being used in the case of 60″ shell, 74 tubes, 16 feet long, and the 66″ shell, 54 tubes, 14 feet long. The number of tubes for a certain diameter of shell is, therefore, limited, for several reasons. Among these is the matter of great area. A line representing a point equidistant between $\frac{1}{7}$ and $\frac{1}{8}$ the total area of grate has proved to be the most efficient, as connected with the ordinary factory chimney. Then there is the matter of uniform and free circulation of the water in the boiler, and having too many tubes prevents the much desired result.

The gases must have a free, and, to a certain extent, rapid passage through the tubes, and must not flow at a velocity less than 132 feet per second, where they leave the boiler.

Then another item that has been overlooked, is the amount of water capacity at the points where the heat is the most intense.

Boilers with tubes too near the bottom and sides of shell *repel* the water, instead of converting it into steam, and in so doing retard the circulation almost entirely at the surfaces, where most of the steam is or should be generated; hence, the low evaporative power of this style.

Besides, the trouble with too many tubes, is the *great* trouble of cleaning; having poor circulation, the deposits collect with alarming rapidity, and in a short time the boiler is as good as ruined; and it cannot be reclaimed, as it is impossible to clean it.

There is just one remedy, and that is to remove some of the tubes.

In support of this theory regarding large tubes, and a limited number of them, we will call to the minds of engineers who practised twenty years ago, the stationary return drop flue boilers, used so extensively at that date in the Eastern States. A 66″ shell boiler would take the gases direct from the furnace through *four flues*, and return them to the front connection through *two flues*. Their average evaporation was 9 to 1; but they were expensive to build, and rather weak in construction at some points.

SHELLS AND TUBES

The shells for all fire tube boilers should be as large as consistent with safety, at the ordinary working pressures, in order to expose as large an area of water level as possible, as this will help to prevent priming.

In regard to heating and grate surface, our proportion is from 32 to 34 square feet of heating surface to one (1) square foot of grate surface or area.

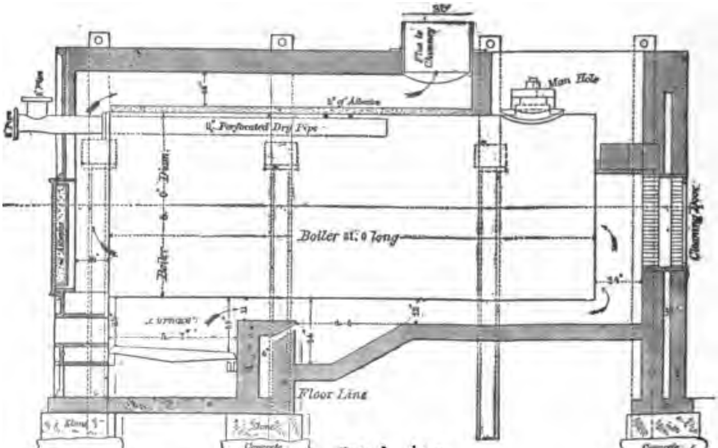
Under no circumstances more than 36 square feet of heating surface to one (1) square foot of grate with natural draught.

The combined tube opening should be no more than one-fifth the total grate area, and not less than one-eighth of the same. The tube opening should *never* with natural draught be subdivided more than sixty times, and fifty-four tubes will generally give the best results.

We should make the length of tubes ***never*** more than fifty diameters, and forty-eight diameters is a good standard for length.

As far as safety is concerned, the plain cylindrical fire tube boiler is not surpassed, as the records of boiler explosions will show upon investigation.

There are many of this style of boilers that have been in constant use for over twenty years, and they are safer to-day than several other kinds entirely new.

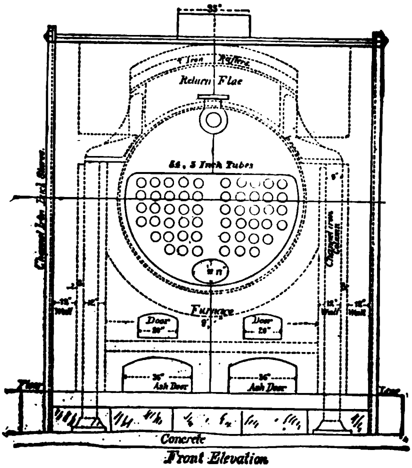


Scale ½ 1 foot

Side Sectional view

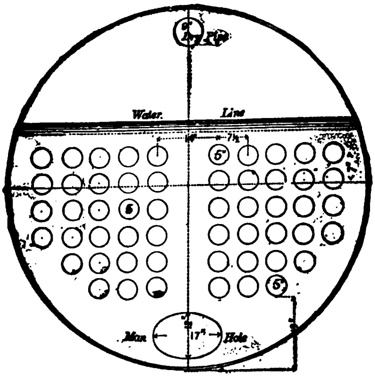
Fig. 34.

Figs. 34, 35, 36 show a working drawing for a 200 H. P. boiler from a design by Wm. H. Hoffman, M. E., Passaic, N. J.



Front Elevation

Fig. 35.



Scale ¼"=1 ft.

Fig. 36.

The tube *opening* is sub-divided fifty-four times. The boiler has no steam dome, a liberal sized dry pipe being substituted for dome.

In the setting it will be noticed that there are *two* side walls on each side of the boiler, with an air space of 3″ between them. The bridge wall is hollow and wide, the space admitting a sufficient quantity of air to the combustion chamber, just behind the bridge wall, to ignite any of the escaping gases.

The heat after leaving the front end of the tubes passes over the top of boiler shell to main chimney flue, there being a thin layer of asbestos on the shell to protect it from any undue rise in temperature. It will be noticed that the brick work does not touch the boiler at any point, they being separated by thin cast iron plates.

The buck-staves are made of wrought channel iron instead of cast iron, with the flat face toward the brick work.

The double wall system will prevent cracking, and is a great source of economy, as the air space becomes a perfect non-conductor.

Two doors are introduced in the fronts for convenience of attending to fires, but there is no partition in the furnace. The boiler can, by the peculiar method of arranging tubes, be thoroughly cleaned inside as well as outside.

Fig. 37 shows the Babcock and Wilcox water tube sectional boiler, which has a world wide reputation as the best boiler of that type yet brought out. For a full description write to Messrs Babcock and Wilcox, (New York City) for a copy of their valuable little book called "Steam." They will send you a copy free of cost.

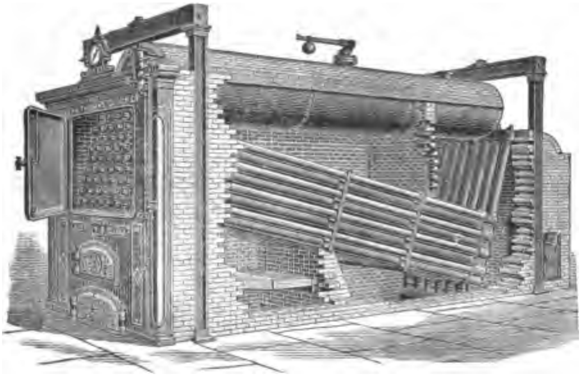


Fig. 37.

Babcock and Wilcox's Sectional Boiler.

This bôiler consists of a series of inclined wrought iron tubes connected by *T* heads, which form the vertical water-channels at each end. The joints are faced by milling them, and then ground so perfectly tight that a pressure of 500 pounds to the square inch is insufficient to produce leakage. No packing is used. The fire is made under the front and higher end of the tubes, and the products of combustion pass up between the tubes into a combustion-chamber under the steam and water drum; hence they pass down between the tubes, then once more up through the space between the tubes, and off to the chimney. The steam is taken out at the top of the steam-drum, near the back end of the boiler. The rapid circulation prevents to some extent the formation of deposits or incrustations upon the heating-surfaces, sweeping them away and depositing them in the mud-drum, whence they are blown out. Rapid circulation of water, as has been shown by Prof. Trowbridge, also assists in the extraction of the heat from the gases, by the presentation of fresh water continually, as well as by the prevention of incrustation.

CHAPTER XI.

UNITED STATES GOVERNMENT GENERAL RULES AND REGULATIONS FOR STEAM (MARINE) BOILERS.

1. Every iron or steel plate intended for the construction of boilers to be used on steam-vessels shall be stamped by the manufacturer in the following manner: At the diagonal corners, at a distance of about four inches from the edges, and at or near the centre of the plate, with the name of the manufacturer, the place where manufactured, and the number of pounds tensile strain it will bear to the sectional square inch.

2. Whenever inspectors shall find a plate of iron or steel with stamps differing as to the tensile strength of the material, they shall rate the tensile strength of the same in accordance with the lowest stamp found thereon.

3. To ascertain the tensile strength of plates, a piece shall be taken from each sheet to be tested, the area of which shall equal one-quarter of one square inch on all plate $\frac{5}{16}$ inch thich and under; and all plate over $\frac{5}{16}$ inch thick the area shall equal the square of its thickness; and the force at which the piece can be parted in the direction of the fiber or grain, represented in pounds avoirdupois—the former multiplied by

four, the latter in proportion to the ratio of its area—shall be deemed the tensile strength per square inch of the plate from which the sample was taken; and should the tensile strength ascertained by the test equal that marked on the plates from which the test-pieces were taken, the plates must be allowed to be used in the construction of marine boilers: *Provided always*, That the plates possess homogeneousness, toughness, and ability to withstand the effect of repeated heating and cooling; but should these tests prove the plates to be overstamped, the lots from which the test-plates were taken must be rejected as failing to have the strength stamped thereon. But nothing herein shall be so construed as to prevent the manufacturers from restamping such plates at the lowest tensile strength indicated by the samples, provided such restamping is done previous to the use of the plates in the manufacture of marine boilers.

4. Local inspectors are required to make and send to the supervising inspectors, with every sample of iron or steel to be tested, their *certificate* that the sample sent for testing was cut from the plate or plates to be used in the boiler designated. And the manufacturer of any boiler to be used for marine purposes shall furnish the inspectors an affidavit in the following form, subscribed to either by himself or authorized agent having superintendence of the construction of such boilers:

AFFIDAVIT OF MANUFACTURER OF MARINE STEAM-BOILERS.

COUNTY OF ———,
State of ———, *ss:*

On this —— day of ———, A. D. 188—, personally appeared before me, a notary public in and for the county of ——— and State of —————, Mr. ——— ———, who, being duly sworn, deposes and says that he is ——— of ——— ———, boiler manufacturer, and that the accompanying samples of ———, manufactured by ——— ———, of ———, were cut from plates stamped ——— T. S., which are to be used in the construction of a marine boiler for ———, and no plate of less tensile strength or quality than herein specified will be used in the construction of said boiler, the dimensions of which will be ———, and of the style known as ———, to be used upon the steamer ———.

——— ———,
———,

Sworn and subscribed to before me this —— day of ———, 188—.

——— ———,
Notary Public.

6. To ascertain the ductility and other lawful qualities, iron of 45,000 pounds tensile strength, and under, shall show a contraction of area of 15 per cent., and each additional 1,000 pounds tensile strength shall show one per cent. additional contraction of area, up to and including 55,000 T. S. Iron of 55,000 T. S. and upwards, showing twenty-five per cent. reduction of area, shall be deemed to have the lawful ductility. All steel plate of one-half inch thickness and under shall show a contraction of area of not less than fifty (50) per cent. Steel plate over one-half inch in thickness shall show a reduction of not less than

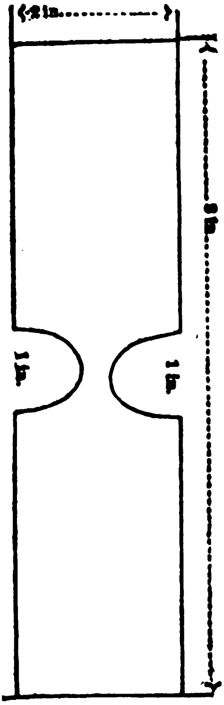
forty-five (45) per cent. *Provided, however*, That steel plate required for repairs to boilers built previous to April 1, 1886, may be used for such repairs when showing a contraction of area of not less than forty (40) per cent.

7. In the following table will be found the width—*expressed in hundredths of an inch*—that will equal, near enough for practical purposes, one-quarter of one square inch of section of the various thicknesses of boiler-plates·

3/16″ × 133	.26 × 96	.35 × 71
.21 × 119	.29 × 86	3/8″ × 67
.23 × 109	5/16″ × 80	7/16″ × 57
1/4″ × 100	.33 × 76	1/2″ × 50

8. The gauge to be employed by inspectors to determine the thickness of boiler-plates, and the widths in the table, will be any standard American gauge furnished by the Treasury Department.

9. All samples intended to be tested on the Riehlé, Fairbanks, or other reliable testing-machine, must be prepared in form, according to the diagram herewith, viz., eight inches in length, two inches in width, cut out at their centers in the manner indicated:



10. All tests made of boiler material must be recorded upon a table of the following form:

TENSILE TESTS OF SAMPLES OF MATERIAL INTENDED TO BE EMPLOYED IN THE CONSTRUCTION OF BOILERS OF STEAM-VESSELS MADE ON —— TESTING-MACHINE.

Date when tests were made.	From whom samples were obtained, and by whom tested.	Material, iron or steel.	Stamp or label on samples, which must be the same as stamps on the material from which they are taken.	Thickness of samples, expressed in hundredths of an inch.	Width of samples, expressed in hundredths of an inch.	Strain at which each sample parted.	Strain per square inch of section.	Reduced thickness.	Reduced width.	Contraction of area—per cent.	Remarks.

(I) PRESSURE ALLOWABLE ON BOILERS OF VARIOUS DIMESIONS BUILT PRIOR TO FEBRUARY 28, 1872.

1. Boilers built prior to February 28, 1872, shall be deemed to have a tensile strength of 50,000 pounds to the sectional square inch, whether stamped or not, and shall be tested under the rule prescribed for boilers inspected under the provisions of section 36 of the act relating to boilers built after the 28th of February, 1872.

* * * * * * *

3. The pressure for any dimension of boilers must be ascertained by the following rule, viz:

Multiply one-sixth ($\frac{1}{6}$) of the lowest tensile strength found stamped on any plate in the cylindrical shell by the thickness—expressed in inches or parts of an inch—of the thinnest plate in the same cylindrical shell, and divide by the radius or half diameter—also expressed in inches—and the sum will be the pressure allowable per square inch of surface for single riveting, to which add twenty per centum for double-riveting.

4. The hydrostatic pressure applied must be in proportion of one hundred and fifty pounds to the square inch to one hundred pounds to the square inch of the steam pressure allowed.

5. Where flat surfaces exist, the inspector must satisfy himself that the bracing and all other parts of the boiler are of equal strength with the shell, and he must also, after applying the hydrostatic test, thoroughly examine every part of the boiler.

6. No braces or stays hereafter employed in the construction of boilers shall be allowed a greater strain than six thousand (6,000) pounds per square inch of section, and no screw stay bolt shall be allowed to be used in the construction of marine boilers in which salt water is used to generate steam. But such screw stay bolts may be used in staying the fire-boxes and

furnaces of such boilers, and not elsewhere, when fresh water is used for generating steam in said boilers.

7. Plates of iron or steel, used in the construction of boilers, extending beyond the cylindrical shell to the front of the boiler over the furnaces, shall extend at least 12 inches below the centre of the shell, and shall not be of less tensile strength or thickness than the adjoining sheets in the cylindrical portions of the shell.

RIVETED FLUES.

8. Flues having a diameter of from 6 inches to 7 inches shall have a thickness of material of not less than .18 of an inch, and if, in the opinion of the inspectors, it is deemed safe, be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .18 .19, .20, .21.

Pressure in pounds, 189, 194, 199, 204.

Flues having a diameter of over 7 inches, and not over 8 inches, shall have a thickness of material of not less than .20 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of the material, as follows:

Thickness of material. .20, .21, .22, .23, .24.

Pressure in pounds, 184, 189, 194, 199, 204.

Flues having a diameter over 8 inches, and not over 9 inches, shall have a thickness of material of not less than .21 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .21, .22, .23, .24, .25, .26.

Pressure in pounds, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 9 inches, and not over 10 inches, shall have a thickness of material of not less than .21 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .21, .22, .23, .24, .25, .26, .27.

Pressure in pounds, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 10 inches, and not over 11 inches, shall have a thickness of material of not less than .22 of an inch, and if, in the opinion of the inspectors, it shall be deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .22, .23, .24, .25, .26, .27, .28, .29.

Pressure in pounds, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 11 inches, and not over 12 inches, shall have a thickness of material of not less than .22 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .22, .23, .24, .25, .26, .27, .28, .29, .30.

Pressure in pounds, 164, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter of over 12 inches, and not over 13 inches, shall have a thickness of material of not less than .23 of an inch, and if, in the opinion of the inspectors, it shall be deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .23, .24, .25, .26, .27, 28, .29, .30, .31, .32.

Pressure in pounds, 159, 164, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 13 inches, and not over 14 inches, shall have a thickness of material of not less than .24 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .24, .25, .26, .27, .28, .29, .30, .31 .32, .33, .34.

Pressure in pounds, 154, 159, 164, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 14 inches, and not over 15 inches, shall have a thickness of material of not less than .25 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .25, .26, .27, .28, .29, .30, .31, .32, .33, .34, .35.

Pressure in pounds, 147, 152, 157, 162, 167, 172, 177, 182, 187, 192, 197.

Flues having a diameter over 15 inches, and not over 16 inches, shall have a thickness of material of not less than .27 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to thickness of material, as follows:

Thickness of material, .26, .27, .28, .29, .30, .31, .32, .33, .34, .35, .36.

Pressure in pounds, 140, 145, 150, 155, 160, 165, 170, 175, 180, 185, 190.

LAP-WELDED FLUES.

9. Lap-welded flues, 16 inches and not less than 7 inches in diameter:

To determine the pressure per square inch allowable on lap-welded flues 18 feet and less in length, multiply the thickness of material in hundredths of an inch by the constant whole number 44, and divide the product by the radius of the diameter of the flue in inches; the quotient will be the pressure allowable.

44 = C, constant.
T = thickness.
R = Radius of diameter in inches.
P = pressure allowable.

Formula: $\frac{C \times T}{R} = P.$

EXAMPLE.

Length of flue, 18 feet; diameter of flue, 14 inches; thickness of material, $\frac{31}{100}$:

Constant, 44.

Then, $\frac{44 \times 31}{7} = 195$lbs., pressure allowable.

For every foot or fraction thereof over 18 feet, deduct three pounds per square inch from the pressure allowable on a flue 18 feet in length or add $\frac{1}{100}$ of an inch to the thickness of material required for a flue 18 feet in length for every 3 feet or fraction thereof over 18 feet.

To determine the thickness of material for any required pressure, multiply the pressure in pounds by the radius of the diameter of the flues in inches, and divide the product by the constant 44; the quotient will be the thickness required.

Formula: $\frac{P \times R}{C} = T.$

EXAMPLE.

Required the thickness of material for a flue 18 feet in length and 14 inches in diameter, requiring a working pressure of 195 pounds to the square inch.

$$\frac{195 \times 7}{44} = 31 +\text{, thickness required.}$$

The thickness of lap-welded flues shall not be less than the product of the constant 2.20 multiplied by the diameter of the flues in inches, which will express the thickness in hundredths of an inch.

Formula: $C \times D = T$, minimum thickness.

EXAMPLE.

Required the least thickness allowable for a lap-welded flue 14 inches in diameter:

$$2.20 \times 14 = \frac{30.80}{100}\text{, least thickness allowable.}$$

Lap-welded flues 7 inches and not over 16 inches in diameter, over 5 feet and not over 10 feet in length, shall be re-enforced by one wrought-iron ring attached externally at the centre of the flue.

Lap-welded flues over 10 feet, and not over 15 feet in length shall have two wrought-iron rings attached to the flue externally, equidistant between the ends of the flue, and there shall be attached one additional ring for every 5 feet or fraction thereof over 15 feet in length.

All such rings shall be good and substantially made, and properly and securely attached to the flues, and shall have a thickness of material of not less than the thickness of the material of the flues, and a width of not less than 2½ inches:

Provided, *however*, Where such flues are made in lengths of not over 5 feet, and fitted one into the other, and substantially riveted, the wrought-iron rings may be dispensed with.

Lap-welded flues seven inches in diameter and less, shall be in accordance with the following table of thicknesses:

Diameter.	Thickness.	Diameter.	Thickness.	Diameter.	Thickness.
Inches.	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*
7	.165	6	.165	5	.148
4½	.134	4	.134	3¾	.120
3½	.120	3¼	.120	3	.109
2¾	.109	2½	.109	2¼	.095
2	.095	1¾	.095	1½	.083
1¼	.072	1	.072		

10. For cylindrical boiler flues over 16 and less than 40 inches in diameter, the following formulas shall be used in determining the pressure allowable:

Let D = diameter of flue, in inches.
$1760 = A$, constant.
T = thickness of flue, in decimals of an inch.
P = pressure of steam allowable, in pounds.
$\frac{1760}{D} = F$, a factor.
$.31 = C$, a constant.

$$\text{Formula: } \frac{F \times T}{C} = P.$$

EXAMPLE.

Given a flue twenty (20) inches in diameter and thirty-seven one hundredths (.37) of an inch in thickness; required the pressure allowable by the inspectors:

$\frac{1760}{D} = F$, a factor; $\frac{1760}{20} = 88 = F$; then supplying values in the formula, $\frac{F \times T}{C} = P$, we have $\frac{88 \times .37}{.31} = P$. Performing the operation indicated, we have $\frac{88 \times .37}{.31} = 105+$ lbs., pressure allowable.

CORRUGATED FURNACE-FLUES.

SEC. 10. The strength of corrugated flues, when used for furnaces or steam-chimneys (corrugation not less than 1½ inches deep), and provided that the plain parts at the ends do not exceed 6 inches in length, and the plates are not less than $\frac{5}{16}$ thick, when new corrugated and practicably true circles, to be calculated from the following formula:

$$\frac{12500}{D} \times T = \text{pressure.}$$

T = thickness in inches.
D = mean diameter in inches.

EXAMPLE.

Given a corrugated flue 40 inches mean diameter, ½ inch thick; required the pressure allowed by inspectors:

$$P = \frac{12500}{D} \times T = \frac{12500 \times .5}{40} = \frac{6250.0}{40} = 156 \text{ pressure.}$$

11. The formulas for cylindrical lap-welded and riveted flues in boilers to be used as furnaces,

which shall be used by inspectors in determining the pressure to be allowed, shall be as follows, viz:

Let D = diameter of flue in inches.
89600=A, constant.
T=thickness of flue in decimals of an inch.
L=length of flue in feet, not to exceed 8 feet.
P=pressure of steam allowable, in pounds.

$$\text{Formula}: \frac{89600 \times T^2}{L \times D} = P.$$

EXAMPLE.

Given, a flue forty (40) inches in diameter, seven (7) feet in length, and five (5) tenths of an inch in thickness; required working pressure to be allowed:

Substituting values in the formula, and performing the operation indicated, we have—

$$P = \frac{89600 \times T^2}{L \times D} = \frac{89600 \times .25}{7 \times 40} = \frac{22400}{280} = 80 \text{ lbs. pressure.}$$

Provided, That if rings of wrought-iron are fitted and riveted properly on, around, and to the flues, in such manner that the tensile strain on the rivets shall not exceed 6,000 pounds per square inch of section, the distance between the rings shall be taken as the length of the flue in the formula.

EXAMPLE.

Given, a flue forty (40) inches in diameter, eight (8) feet long, and five (5) tenths of an inch in thickness, having one ring at the middle of its length; required the pressure allowable by the inspectors.

Substituting values in the formula, and performing the operation, we have—

$$P = \frac{89600 \times T^2}{L \times D} = \frac{89600 \times .25}{4 \times 40} = \frac{22400}{160} = 140 \text{ lbs. pressure.}$$

12. The feed-water shall not be admitted into any boiler at a temperature less than one hundred degrees Fahrenheit for low-pressure boilers, and one hundred and eighty for high-pressure boilers.

13. Whenever steamers use a pressure upon their boilers exceeding sixty pounds to the square inch, they shall be inspected as high-pressure steamers and designated as such.

14. Vertical tubular boilers shall not be used on steamers navigating the Red River of the North and rivers whose waters flow into the Gulf of Mexico, unless the water-line is 2 inches above the upper end of the tubes and fire-line.

15. All steamers navigating rivers, having boilers externally heated, shall have a clear space of not less than six inches between the boilers and wood-work on either side, and four inches on the top of said boilers.

16. All steamers navigating the ocean, sounds, lakes, bays, and rivers, the boilers of which shall be internally heated, shall have a clear space of at least four inches on either side, and at the top not less than two inches clear space above the covering of the boilers.

17. All boilers hereafter placed in steamers shall have a clear space of at least eight inches between the under side of the cylindrical shell and the floor or keelson.

All man-holes for the shell of boilers shall have an opening not less in diameter than 11 by 15 inches in the clear, except that boilers less than 34 inches diameter of shell have an opening in the clear, in man-holes, of not less than 9 by 14⅞ inches; all boiler shells between 34 and and 38 inches diameter, an opening not less than 9 by 16 inches, and all boiler shells between 38 and 48 inches in diameter, an opening not less than 11 by 15⅞ inches.

18. All wood-work or other ignitible substance, approaching within two inches of the boiler, shall be suitably sheathed with metal, so adjusted to permit a free circulation of air between the sheathing and the ignitible surface.

19. All boilers shall have a clear space at the back and ends thereof of two feet opposite the back-connection door. Slip-joints in steam-pipes shall, in their working parts, when the steamer is to be employed in navigating salt water, be made of copper or composition.

20. There shall be fastened to each boiler a plate containing the name of the manufacture of the material, the place where manufactured, the tensile strength, the name of the builder of the boiler, when and where built.

21. Every sea-going steamer carrying passengers shall be supplied with an auxiliary or donkey boiler of sufficient capacity to work the fire-pumps.

22. All steamers shall have inserted in their boilers plugs of Banca tin, at least one-half inch in diameter at the smallest end of the internal opening, in the following manner, to wit: Cylinder boilers with flues shall have one plug inserted in one flue of each boiler; and also one plug inserted in the shell of each boiler from the inside, immediately before the fire line, and not less than four feet from the forward end of the boilers. All fire-box boilers shall have one plug inserted in the crown of the back connection, or in the highest fire service of the boiler. All upright tubular boilers used for marine purposes shall have a fusible plug inserted in one of the tubes at a point at least two inches below the lower gauge-cock, and said plug may be placed in the upper head sheet when deemed advisable by the local inspectors. All fusible plugs, unless otherwise provided, shall have an external diameter not less than that of a one-inch gas or steam pipe screw-tap, except when such plugs shall be used in the tubes of upright boilers; plugs may be used with an external diameter of not less than that of a three-eighths of an inch gas or steam pipe screw-tap, said plugs to conform in construction with plugs now authorized

to be used by this Board; and it shall be the duty of the inspectors to see that these plugs are filled with Banca tin at each annual inspection.

23. All steamers having one or two boilers shall have three suitable gauge-cocks in each boiler. Those having three or more boilers in battery shall have three in each outside boiler and two in each remaining boiler in the battery; and the middle gauge-cocks in all boilers shall not be less than four inches above the top of the flues, tubes, or crown of the fire-box.

24. Lever safety-valves to be attached to marine boilers shall have an area of not less than one square inch to two square feet of the grate surface in the boiler, and the seats of all such safety-valves shall have an angle of inclination of 45 degrees to the center line of their axis.

The valves shall be so arranged that each boiler shall have one separate safety-valve, unless the arrangement is such as to preclude the possibility of shutting off the communication of any boiler with the safety valve or valves employed. This arrangement shall also apply to lock-up safety-valves when they are employed.

Any spring-loaded safety-valves constructed so as to give an increased lift by the operation of steam, after being raised from their seats, or any spring-loaded safety-valve constructed in any other manner, or so as to give an effective area equal to that of the aforementioned spring-loaded safety-valve, may be used in lieu of the common lever-weighted valve on all boilers on all boilers on steam vessels, and all such spring-loaded safety-valves shall be required to have an area of not less than one square inch to 3 square feet of grate-surface of the boiler, and each spring-loaded valve shall be supplied with a lever that will raise the valve from its seat a distance of not less than that equal to one-eighth the diameter of the valve-opening, and the seats of all such safety-valves shall have an angle of inclination to the center-line of their axis of 45 degrees. But in no case shall any spring-loaded safety-valve be used in lieu of the lever-weighted safety-valve, without first having been approved by the Board of Supervising Inspectors.

The first paragraph of this section applies to valves constructed in material, workmanship and principle according to the drawings for a safety-valve printed with these rules, and all common lever safety-valves to be hereafter applied to the boilers of steam vessels must be so constructed.

When this construction of a safety-valve is applied to the boilers of steamers navigating rough waters, the link may be connected direct with the spindle of the valve: *Provided always*, That the fulcrum or points upon which the lever rests are made of steel, knife or sharp edged, and hardened; in this case the short end of the lever should be attached directly to the valve-casing.

In all cases the link requires but a slight movement, not exceeding one-eighth of an inch.

All the points of bearing on lever must be in the same plane.

The distance of the fulcrum must in no case be less than the diameter of the valve-opening.

The length of the lever should not exceed the distance of the fulcrum multiplied by ten.

The width of the bearings of the fulcrum must not be less than three-fourths (¾) of one inch.

The length of the fulcrum link should not be less than four inches.

The lever and fulcrum link must be made of wrought-iron or steel, and the knife-edged fulcrum points and bearings for the points must be made of steel and hardened.

The valve, valve-seat, and bushings for the stem or spindle must be made of composition (gun-metal) when the valve is intended to be attached to a boiler using salt water; but when the valve is to be attached to a boiler using fresh water, and generating steam of a high pressure, the parts named, with the exception of the bushings for the spindle, may be made of cast-iron.

The valve must be guided by its spindle, both above and below the ground seat and above the lever, through supports either made of composition (gun-metal) or bushed with it.

The spindle should fit loosely in the bearings or supports.

When the valve is intended to be applied to the boilers of steamers navigating rough waters, the fulcrum-link may be connected directly with the spindle of the valve; providing always, that the knife-edged fulcrum points are made of steel and hardened, and that the vertical movement of the valve is unobstructed by any lateral movement.

In all cases the weight must be adjusted on the lever to the pressure of steam allowed in each case by a correct steam-gauge attached to the boiler. The weight must then be securely fastened in its position and the lever marked, for the purpose of facilitating the replacing of the weight should it be necessary to remove the same; and in no case shall a line or any other device be attached to the lever or weight except in such a manner as will enable the engineer to raise the valve from its seat.

Donkey boilers used on all steam vessels for driving pumps, hoisting engines, electric lights, or other purposes, must be inspected the same as the main steam-boilers, and supplied with water and steam gauges, and the safety-valves must comply with the same regulations as the main boilers.

The area of all openings in boilers and connections leading from boilers to safety-valves, both the lever and spring-loaded valves, used on marine boilers, shall not be less than the area of the valve used in said safety-valve.

25. All steam-gauges heretofore in use on steamers shall be admissible by the inspectors, and other steam-gauges hereafter made, of equal merit, shall be allowed.

26. All boilers or sets of boilers shall have attached to them at least one gauge that will correctly indicate a pressure of steam equal to 80 per cent. of the hydrostatic applied by the inspectors.

27. The appliances in use on steamers constructed prior to the 28th of February, 1872, for determining the height of water in the boilers shall be considered reliable low-water gauges.

28. There must be means provided in all boilers using the "low-water gauges," which are operated by means of a float inside the same, to prevent the float from getting into the steam pipe.

29. In applying the hydrostatic test to boilers with a steam-chimney, the test-gauge should be applied to the "water-line" of such boilers.

30. All horizontal cylindrical boilers used on steamers navigating the waters flowing into the Gulf of Mexico shall be provided with a reliable low-water gauge.

31. The hydrostatic test applied to the boilers of towing freight-boats on the Mississippi River and its tributaries shall be in the proportion of one hundred and fifty (150) pounds to one hundred pounds working steam pressure allowed;

and the inspectors shall test all such boilers on said steamers for the amount of steam allowed.

STEAM PRESSURE ALLOWED ON BOILERS.

[32.] Thickness of iron.	34 inches diameter.	36 inches diameter.	38 inches diameter.	40 inches diameter.	42 inches diameter.	44 inches diameter.	46 inches diameter.
Inches.	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*
.19	140.82	133.	126.	119.70	114.	108.81	104.08
.20	148.23	140.	132.63	126.	120.	114.54	109.56
.21	155.64	147.	139.26	132.30	126.	120.27	115.04
.22	163.05	154.	149.50	138.60	132.	126.	120.52
.23	170.47	161.	152.52	144.90	138.	131.72	126.
.24	177.88	168.	159.15	151.20	144.	137.45	131.47
.25	185.29	175.	165.79	157.50	150.	143.18	136.95
.26	192.70	182.	172.42	163.80	156.	148.80	142.43
.27	200.11	189.	179.05	170.10	162.	154.63	147.91
.28	207.53	191.	185.68	176.40	168.	160.36	152.52
.29	214 94	203.	192.31	182.70	174.	166.04	158.87
.30	222.35	210.	198.94	189.	180.	171.81	164.34
.31	229.76	217.	205.57	195.30	186.	177.54	169.82

LICENSED OFFICERS.

1. Before an original license is issued to any person to act as a master, mate, pilot, or engineer, he must personally appear before some local board or a supervising inspector for examination; but upon the renewal of such license, when the distance from any local board or supervising inspector is such as to put the person holding the same to great inconvenience and expense to appear in person, he may, upon taking the oath of office before any person authorized to administer oaths, and forwarding the same, together with the license to be renewed and Government fee, to the local board or supervising inspector of the district in which he resides or is employed, have the same renewed by the said

inspectors, if no valid reason to the contrary be known to them; and they shall attach such oath to the stub end of the license, which is to be retained on file in their office. And inspectors are directed, when licenses are completed, to draw a broad pen and red ink mark through all unused spaces in the body thereof, so as to prevent, so far as possible, illegal interpolation after issue.

2. The classification of engineers on the lakes and seaboard shall be as follows:

Chief engineers of ocean steamers.

Chief engineers of lake, bay, and sound steamers.

Chief engineers of river steamers.

First assistant engineers.

Second assistant engineers.

Third assistant engineers.

Special engineers.

All steamers of over one hundreds tons burden shall carry at least one chief engineer.

First assistant engineers may act as first assistants on any steamer.

Second assistant engineers may act as first assistants on steamers of seven hundred and fifty tons and under, and second assistants on any steamer.

Third assistants may act as second assistants on steamers of 750 tons and under, and third assistants on any steamer.

Special engineers may be assigned to act in any capacity for which they are qualified on steamers of one hundred tons and under.

Inspectors may designate upon the certificate of any chief engineer the tonnage of the vessel on which he may act, and they may also designate any assistant engineer as special engineer on steamers of one hundred tons or under, and may restrict an engineer to a particular vessel.

3. Engineers on high-pressure steamers navigating rivers shall be designated as chief engineers (H. P.), assistant engineers (H. P.), and special engineers (H. P.).

Assistant engineers may act as chief engineers on high-pressure steamers of one hundred tons burden and under, of the class and tonnage, or particular steamer for which the inspectors, after a thorough examination, may find them qualified. In all cases where an assistant engineer is permitted to act as first [chief] engineer, the inspectors shall state on the face of his certificate of license the class and tonnage of steamers, or the particular steamer, on which he may so act.

4. It shall be the duty of an engineer, when he assumes charge of the boilers and machinery of a steamer, to forthwith thoroughly examine the same, and if he finds any part thereof in bad condition, caused by neglect or inattention on the part of his predecessor, he shall immediately report the facts to the local inspectors of the dis-

trict, who shall thereupon investigate the matter, and if the former engineer has been culpably derelict of duty, they shall suspend or revoke his license.

5. No original license shall be issued to any person to act as engineer, except for special license on small pleasure steamers, who cannot read and write, or who has not served at least three years in the engineer's department of a steam-vessel, or as a regular machinist in a machine works; provided that any person who has served for a period of three years as a locomotive or a stationary engineer may be licensed to act as engineer on steam vessels after having had not less than one year's experience in the engineer's department of a steam vessel.

CHANGES IN STEAMBOAT RULES.

Amendments Made by the Supervising Inspectors and Approved.

Section 5, rule 5, has been amended so that no person shall receive an original license as engineer who has not served at least three years in the engineer's department of a steam vessel, except locomotive and stationary engineers, and a regular steam engine machinist; and no person shall receive a license who is not able to determine the weight necessary to be placed on the lever of a safety-valve to withstand any given pressure of steam in a boiler. By section 18 the grade of an engineer or pilot shall not be raised during the term for which the license was granted except by consent of the board that granted the license.

CHAPTER XII.

TO FIND THE SIZES (DIAMETER) OF CYLINDERS FOR COMPOUND ENGINES—HORSE POWER REQUIRED—BOILER PRESSURE AND SPEED OF PISTON BEING GIVEN.

TAKE for an example say: horse power = 100; initial pressure = 60 pounds (gauge); and piston speed 400 feet. First find the number of times the steam is expanded.

Rule: Divide the absolute initial pressure by the terminal pressure. The quotient will be the number of times the steam is expanded.

Thus, $\frac{60 + 15}{15} = 5$, which is the number of expansions in both cylinders.

The area of the low pressure cylinder is the same as the area of the cylinder of a simple cylinder engine working under the same expansion: hence find the mean pressure as for a simple cylinder by the following rule:

Find the hyperbolic logarithm for the number representing the expansion, to which add 1, and multiply the sum by the terminal pressure.

The product will be the mean pressure.

The hyp. log. of 5 is 1.609, and 1.609 + 1 = 2.609, and 2.609 × 15 = 39 +, which is the mean pressure in pounds per square inch.

The area of cylinder is found by the following rule:

Multiply 33,000 by the number of horse power, and divide the product by the piston speed, multiplied by the mean pressure. The quotient will be the area of the cylinder in inches.

Thus, $\frac{33000 \times 100}{400 \times 39} = 211.5+$, corresponds to a diameter of about 16½″.

The diameter of high pressure cylinder should be such as to equalize the maximum strains on the two pistons; in other words, the expansion should be the same in each cylinder.

Hence, to find the area of high pressure cylinder:

Divide the area of the low pressure cylinder by the square root of the number of times the steam is expanded.

The quotient will be the area of the high pressure cylinder in inches.

Thus, $\frac{211\frac{1}{2}}{\sqrt{5}} = 94.5+$, which corresponds to a diameter of about 11″.

To find the point of cut-off in a high pressure cylinder:

Divide the area of the high pressure cylinder by the area of the low pressure cylinder.

The quotient will be the fraction of stroke which shouid be made before steam is cut off.

Thus, $\frac{94.5}{211.5} = .446+$, which is the point of cut-off in the

high pressure cylinder, measuring from the commencement of the stroke.

For the problem submitted we have: Diameter of low pressure cylinder, 16½″, diameter of high pressure cylinder, 11½″.

Point of cut-off in high pressure cylinder, .446 of stroke.

This consideration of the subject is purely theoretical, and expresses what ought to be rather than what is.

For instance, an allowance must be made for back pressure, loss of pressure between the cylinders, friction of engine, etc.

No absolute rule can be given for this, nor even an approximation, unless all the details of construction are known.

In practice the dimensions of cylinders, as found above, would be increased to an extent which corresponded to the individual builder's taste or experience.

CHAPTER XIII.

TO FIND THE MEAN PRESSURE.

Divide the length of the stroke by the length of the space into which the steam is admitted; find in the table the logarithm of the number nearest to the quotient, to which add 1, the sum is the ratio of the grain; then find the terminal pressure by dividing the initial pressure by the proportion of the stroke during which the steam is admitted, and multiply it by the logarithms + 1, found as above; the product will be the mean pressure through the stroke.

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.
1.25	.22314	5.	1.60943	9.5	2.25129
1.5	.40546	5.25	1.65822	10.	2.30258
1.75	.55961	5.5	1.70474	11.	2.39789
2.	.69314	5.75	1.74919	12.	2.48490
2.25	.81093	6.	1.79175	13.	2.56494
2.5	.91629	6.25	1.83258	14.	2.63905
2.75	1.01160	6.5	1.87180	15.	2,76805
3.	1.09861	6.75	1.90954	16.	2.77258
3.25	1.17865	7.	1.94591	17.	2.83321
3.5	1.25276	7.25	1.98100	18.	2.89037
3.75	1.32175	7.5	2.01490	19.	2.94443
4.	1.38629	7.75	2.04769	20.	2.99573
4.25	1.44691	8.	2.07944	21.	3.04452
4.5	1.50507	8.5	2.14006	22.	3.09104
4.75	1.55814	9.	2.19722		

Example 1. Suppose the length of the stroke to be 48 inches, the initial pressure to be 40 pounds per square inch, and the steam to be cut off at 12 inches of the stroke, what will be the mean pressure?

$48 \div 12 = 4$. Hyp. log. of 4, $= 1.38629 + 1 = 2.38629$. Then $40 \div 4 = 10 \times 2.38629 = 23.8629$ pounds, the mean pressure required.

Example 2. Suppose the length of the stroke to be $36''$, initial pressure to be 50 pounds per square inch, and the steam to be cut off at $9''$ of the stroke, what will be the average pressure?

$36 \div 9 = 4$. Hyp. log. of $4 = 1.38629 + 1 = 2.38629$. Then $50 \div 4 = 12.5 \times 2.38 = 29.75$, mean pressure required.

This is correct without taking the clearance into account.

With the clearance added, the mean pressure would be slightly greater.

Or 2d, by means of the following table. The pressures per table are 15 pounds greater than the pressure that would be shown by a correct steam-gauge.

In the "Average Pressure" no deduction is made for back-pressure. Deducting sixteen pounds for a non-condensing engine and, say, six pounds for a condensing engine, we have the mean effective pressure.

For example, it is required to find the power of an engine twenty inches diameter of cylinder, running at a piston speed of 600 feet per minute, admitting steam of seventy pounds pressure, *by gauge*, to the cylinder, and cutting off at a quarter stroke. Add fifteen to seventy to find the absolute pressure; then against eighty-five and under a quarter cut-off the average pressure is found to be 50.65 pounds. This, less sixteen pounds, gives 34.65 pounds as the mean effective pressure non-condensing, and less six pounds, 44.65 pounds as the mean effective pressure condensing.

MEAN PRESSURE OF STEAM AT DIFFERENT RATES OF EXPANSION.

Initial pressure in pounds per square inch.	Average pressure in pounds per square inch for the whole stroke.							
	Points in the stroke at which steam is cut off.							
	$\frac{1}{10}$	$\frac{2}{10}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{3}{8}$	$\frac{4}{10}$	$\frac{5}{10}$	$\frac{6}{10}$
40	13.21	20.87	23.86	26.22	29.67	30.66	33.86	36.14
45	14.86	23.48	26.84	29.73	33.38	34.89	38.09	40.66
50	16.51	26.09	29.82	33.03	37.07	38.32	42.32	45.18
55	18.12	28.57	32.86	36.67	40.83	42.08	46.47	49.91
60	19.79	31.02	35.77	39.23	44.49	45.98	50.73	54.41
65	21.49	33.84	38.71	42.98	48.35	49.89	54.98	58.96
70	23.18	36.43	41.73	46.22	52.00	53.52	59.07	63.25
75	24.72	39.00	44.82	49.53	55.73	57.36	63.38	68.00
80	26.41	41.66	47.75	52.88	59.41	61.13	67.47	72.45
85	28.02	44.08	50.65	56.14	63.17	65.00	71.84	77.00
90	29.69	46.89	53.73	59.43	66.94	68.83	76.00	81.47
95	31.34	49.37	56.62	62.81	70.52	72.68	80.16	86.00
100	33.03	52.00	59.63	66.01	74.23	76.47	84.37	90.57
110	36.41	57.46	65.87	72.97	81.82	84.12	93.00	99.95
120	39.77	62.50	71.84	79.44	89.10	91.98	101.44	108.63
130	43.01	67.85	77.49	86.00	96.55	99.47	110.00	117.89
140	46.32	73.00	83.38	92.51	104.00	107.21	118.36	126.77
150	49.73	78.11	89.12	99.04	111.63	114.56	126.89	136.00

INDICATED HORSE-POWER FOR EACH POUND MEAN EFFECTIVE PRESSURE, WITH DIFFERENT DIAMETER AND SPEEDS OF PISTONS.

Diam.of cylinder.	Speed of piston in feet a minute.									
	240	300	350	400	450	500	550	600	650	750
In										
11	.691	.864	1.008	1.[illegible]	1.296	1.44	1.584	1.728	1.872	2.160
12	.820	1.025	1.195	1.300	1.540	1.708	1.880	2.050	2.222	2.564
13	.964	1.206	1.407	1.608	1.809	2.01	2.211	2.412	2.613	3.015
14	1.119	1.398	1.631	1.864	2.097	2.331	2.564	2.797	3.029	3.495
15	1.285	1.606	1.873	2.131	2.409	2.677	2.945	3.212	3.479	4.004
16	1.461	1.827	2.131	2.436	2.741	3.045	3.349	3.654	3.958	4.567
17	1.643	2 054	2.396	2.739	3.081	3.424	3.766	4.108	4.450	5.135
18	1.849	2.312	2.697	3.083	3.468	3.854	4.239	4.624	5.009	5.78
19	2.061	2.577	3.006	3.436	3.865	4.295	4.724	5.154	5.583	6.442
20	2.292	2.855	3.331	3.807	4.285	4.759	5.234	5.731	6.186	7.138
21	2.518	3.148	3.672	4.197	4.722	5.247	5.771	6.296	6.820	7.869
22	2.764	3.455	4.031	4.607	5.183	5.759	6.334	6.911	7.486	8.638
23	3.021	3.776	4.405	5.035	5.664	6.294	6.923	7.552	8.181	9.44
24	3.289	4.111	4.797	5.482	6.167	6.853	7.538	8.223	8.908	10.279
25	3.569	4.461	5.105	5.948	6.692	7.436	8.179	8.923	9.566	11.053
26	3.861	4.826	5.630	6.435	7.239	8.044	8.848	9.652	10.456	12.065
27	4.159	5.199	6.066	6.932	7.799	8.666	9.532	10.399	11.265	12.998
28	4.477	5.596	6.529	7.462	8.395	9.328	10.261	11.193	12.125	13.991
29	4.805	6.006	7.007	8.008	9.009	10.01	11.011	12.012	13.013	15.015
30	5.141	6.426	7.497	8.568	9.639	10.71	11.781	12.852	13.923	16.065
31	5.486	6.865	8.001	9.144	10.287	11.43	12.573	13.716	14.866	17.145
32	5.846	7.308	8.526	9.744	10.962	12.18	13.398	14.616	15.834	18.270
33	6.216	7.770	9.065	10.360	11.655	12.959	14.245	15.54	16.835	19.425
34	6.59	8.238	9.611	10.984	12.357	13.73	15.103	16.476	17.849	20.595
35	6.993	8.742	10.199	11.656	13.113	14.57	16.027	17.484	18.941	21.855
36	7.401	9.252	10.794	12.336	13.878	15.42	16.962	18.504	20.046	23.130
37	7.819	9.774	11.403	13.032	14.861	16.29	17.919	19.548	21.177	24.435
38	8.246	10.308	12.026	13.744	15.462	17.18	18.898	20.616	22.334	25.770
39	8.648	10.86	12.67	14.48	16.29	18.1	19.91	21.62	23.53	27.15
40	9.139	11.424	13.328	15.232	17.136	19.04	20.944	22.848	24.752	28.560
41	9.604	12.006	14.007	16.008	18.009	20.00	22.011	24.012	26.013	30.015
42	10.065	12.594	14.693	16.792	18.901	20.99	23.089	25.188	27.287	31.485
43	10.56	13.20	15.4	17.6	19.8	22.0	24.2	26.4	28.6	33.0
44	11.046	13.818	16.121	18.424	20.727	23.03	25.333	27.636	29.939	34.545
45	11.563	14.454	16.863	19.272	21.681	24.09	26.399	28.908	31.317	36.135
46	12.086	15.128	17.626	20.144	22.662	25.18	27.698	30.216	32.754	37.770
47	12.614	15.768	18.396	21.024	23.652	26.28	28.908	31.536	34.164	39.420
48	12.846	16.446	19.187	21.928	24.669	27.41	30.151	32.152	35.633	41.115
49	12.913	17.142	19.999	22.856	25.713	28.57	31.427	34.284	37.141	42.855
50	14.28	17.85	20.825	23.8	26.775	29.75	32.725	35.7	38.675	44.625
51	14.832	18.54	2, 665	24.76	27.855	30.95	34.045	37.08	40.205	46.425
52	15.437	19.296	22.512	25.728	28.944	32.16	35.376	38.592	41.808	48.240
53	16.041	20.052	23.394	26.736	30.078	33.42	36.762	40.104	43.446	50.13
54	16.650	20.82	24.29	27.76	31.23	34.7	38.17	41.64	45.11	52.05
55	17.275	21.594	25.193	28.792	32.391	35.99	39.589	43.188	46.787	53.985
56	17.909	22.386	26.117	29.848	33.579	37.31	41.041	44.772	48.503	55.965
57	18.557	23.196	27.062	0.928	34.794	38.66	42.526	46.392	50.258	57.99
58	19.214	24.018	28.021	32.024	36.027	40.03	44.033	48.036	52.039	60.045
59	19.902	24.852	28.994	33.136	37.278	41.42	45.562	49.704	53.846	62.13
60	20.558	25.698	29.981	4.264	38.547	42.83	47.113	51.396	55.679	64.245

The table gives the horse power for ***each pound*** mean effective pressure.

The horse power of a 20″ cylinder, 600 feet piston speed, is found to be 5.731 for each pound mean effective pressure. Multiplying this by 34.65 gives 198 horse power non-condensing, and by 44.65, 255 horse power condensing.

CHAPTER XIV.

THE TRENTON STEAM ENGINE. AUTOMATIC CUT-OFF.

Introduction.

THE Phœnix Iron Company, (Trenton, N. J.) after several years of experiment and close observation, have succeeded in creating four classes of steam engines, applicable to all the varying requirements to which prime motors may be subject.

It is their aim to furnish an engine designed and constructed in accordance with modern theory and practice, and containing all the elements of economy, such as:

I. STRENGTH.
II. SIMPLICITY.
III. DURABILITY AND LEAST WEAR.
IV. ECONOMY IN THE USE OF STEAM.
V. REGULARITY OF SPEED.
VI. FITNESS FOR ITS WORK.

I. Strength.

Their engines are designed to resist the greatest strains arising from any manner of loading. All parts are so proportioned as to insure the greatest durability, and to prevent tremors or strains.

They have deviated from the proportions and forms to be met with in other standard engines; a difference made apparent by examination of the cuts and description in the following pages.

II. Simplicity.

They have made it a point of the greatest importance to build an engine containing the least number of parts, and the simplest elements of mechanical movements attainable, in construction and design, such as form of frame or bed, piston head, packing rings, cross head, guides, connecting-rod ends, valve gear, valves and regulating appliances.

To these parts of their engine we invite special attention.

III. Durability and Least Wear.

Whilst the rapid development of steam engineering of recent years had led to a higher rate of speed, it has also demonstrated that, practically, there is a limit; a limit which, if exceeded, leads to an early destruction of the engine, after annoying halts and expensive repairs.

Before deciding upon the speeds for their engines, they carefully observed the effects, upon engines, of high speeds which had been adopted, or rather attempted, under the now-proved erroneous impression that the rate of speed of a perfectly constructed engine was unlimited.

They came to the conclusion that there exist so many unsurmountable practical difficulties, that very high speed is undesirable, impracticable, and poor economy.

The least variation in the line of the engine, a slight settlement of the foundation, unequal wear of a bearing, or the failure of unguents to flow but for a moment, will cause parts to heat, wear and ultimately fracture.

We have yet to see the high-speed engine that has continued to run at the velocity originally intended.

Taking all these contingencies into consideration, they have been led to adopt speeds greater than previously practiced, but less than those advocated by some contemporary builders.

Their speeds are based upon experience, with due regard to economy, comprising durability, continuous work, and the least amount of fuel.

The main feature of their Balanced Valve is the peculiarity of its design, being cylindrical, very light, absolutely balanced, having a large wearing surface, and a combined sliding and rotary motion; thus insuring a continual change in the surfaces of contact, and thereby avoiding the cutting or grinding away of a soft spot by a hard one from the continuous action over the same track.

The sliding motion, which is positive, exhausts and cushions the steam; whilst the rotary motion, which is variable, depending upon the point of cut-off or ratio of expansion, admits the steam to and cuts it off from the cylinder.

These valves have proved their excellence and superiority over all others; having been tried on an engine now running three years, night and day continuously, in a rolling mill, and being as perfect now as when first started.

IV. Economy in the Use of Steam.

An element of vital importance in a steam engine is, of course, the distribution of steam in the cylinder. It embraces the least amount of clearance or dead space, consistent with the smooth running of the engine; prevention of the loss of heat from the steam in the cylinder; an initial steam pressure in the cylinder, equal to the boiler pressure; a minimum back pressure, rapid action of the admitting and cutting-off edges of the valve; a proper amount of cushioning; and last, but not least, a fitness of the engine for its work; its size and the character of the valve gear.

The least amount of clearance depends upon the careful designing of the engine, in that the valve seats shall be close to the cylinder, and the dead space between the valve seats and the end of the piston-head reduced to a minimum.

without restricting the area of the ports and the necessary space between the piston-head and the head of the cylinder.

These valves, but two in number, admitting and cutting off the steam through one port instead of two at each end of the cylinder, have enabled them to greatly diminish this clearance, by saving the volume of steam required to fill one port at each stroke of the engine; a most important factor in their favor.

All parts of the cylinders of these engines are covered with the most effective non-conducting material, to insure the retention of heat and prevent liquefaction.

To attain a high initial steam pressure in the cylinders, the ports and steam passages of these engines are proportioned for a velocity of the steam to fill the cylinders at the maximum speed of the engines, of not over 80 feet per second; at which velocity, the loss from friction in the steam pipes and passages is at a minimum, all things considered.

The steam edges of these valves for large engines being in the form of a helix, and having a combined sliding and rotary motion, are made to open doubly as quick as those with transverse or longitudinal edges only; the velocity of the sliding action being equal to that obtained in all other engines, to which they add the speed or motion obtained from the rotary action; so that they get full ports at $\frac{3}{16}$ the stroke of the piston, or double the opening usually attainable, a result of the greatest importance.

To obtain the same result, viz.: the quick admission of steam through the valves in their Class C and D engines, they use their patented, double-ported valves, more fully described further on.

V. Regularity of Speed.

One of the most essential features of a good engine is regularity of speed under varying intensities of load.

They have succeeded in the construction of two different kinds of governors, applicable to their different classes of engines, absolutely isochronous, and changing the ratio of expansion or point of cut-off, with the least change in the load.

One is for their Class A engines, with a releasing or trip gear; and the other (a centrifugal crank-shaft governor), for their Class B, C and D engines, with positive-motion valves; the throw off valves being changed, as also the angular advance of the eccentric, without the least change in the lead of the steam edge.

Both governors may be set to any desired speed, and are not affected by the friction from the valve gear.

By actual test, these engines have shown a regularity of speed varying only 3 per cent. in two revolutions; the variation of load being 80 per cent.

VI. Fitness of an Engine for its Work.

It is desirable to call the attention of steam users to the great importance of choosing an engine of the proper size and character of valve gear, suitable for the work it is intended for.

It would be folly to purchase a fixed cut-off engine for greatly varying loads; so also would it be poor economy to apply to a small steady load, an expensive automatic cut-off engine, or one so large that the ratio of expansion becomes excessive and exceeds the limits of economy.

Each case requires its special engine of proper proportions.

Medium loads, with varying intensity, requiring high rotative speed, should have a valve gear cutting off the steam at different points of the stroke, automatic in its action, and with a positive motion; the releasing or trip action, depending upon gravity or springs for imparting motion to the valves, would be impracticable.

For different and varying needs of steam users, are recommended:

CLASS A.	For moderately high speeds, varying loads and great power.
CLASS B.	For high rotative speeds, varying loads and great power.
CLASS C AND D.	For high rotative speeds, and small power.

All four classes are, of course, applicable to any speed and load.

To steam users having objections to the

modern tendency toward increased rates of speed, and who do not object to a comparatively greater first cost of engine, we recommend their Class A engines, which are adapted to lower rotative speeds than Class B, C or D.

TRENTON ENGINE—CLASS A. AUTOMATIC CUT-OFF.

General Description.

This engine is of the automatic cut-off type, with balanced valves, actuated by a trip or releasing gear.

It is designed for variable heavy work.

The cylinder and its base are cast in one piece; the steam chest, containing the two balanced valves, being bolted thereto.

The frame or bed is cast together with the pillow block or main bearing and guides, and bolted to the cylinder through flanges and head.

The crank pin and cross-head pin are of steel, also the piston rod, and all of very large proportions. The crank pin and cross-head boxes are of phosphor-bronze; also the sliding pieces of the cross-head and the side bearings of the pillow block.

The valves are circular in cross section, accurately ground to a true cylinder to closely fit the bore of the steam chest, which is very carefully reamed after boring.

They fit loosely upon the valve stem, and are balanced endwise by means of phosphor-bronze discs fastened to the valve stem, so that the pressure of steam cannot create any friction or wear, thus insuring an easy motion.

The steam edges of the valves are in the form of a helix, so that the steam is admitted by the simultaneous sliding and circular motion, which tends to open the ports quickly.

The steam ports are widened towards the exhaust ends, so that the distance measured around the bore of the steam chest corresponds to the width of the exhaust port, which equals the diameter of the cylinder.

The valves, which are indepenedent of each other as to circular motion, are rotated alternately by the reciprocating motion of the eccentric, through the eccentric rod, rock-arm, and the two vertical valve stems which pass through the steam chest.

The lower ends of these stems are connected to pistons working in dash-pots, and the upper ends to the releasing or trip gear.

At about the middle of these stems are fastened yokes carrying phosphor-bronze boxes, through which the crank stems pass, and which are fastened to and rotate the valves through the vertical motion of the stems.

The vertical motion of the stems is imparted by means of the above-mentioned rock-arm; the horizontal arms of which carry phosphor-bronze boxes, which slide in yokes moving up and down in cast-iron guides.

The cast-iron guides, to which the vertical valve stems are attached, are supplied with triggers, against which, at each upward motion, the yokes in the guides press; thus causing the motion of the rock-arm to be imparted to the vertical stems.

When the triggers come in contact with the levers, which are adjusted to different positions by the sliding plates or wedges, which again are dependent, for their position, upon the speed of the governor, they are thrown out and the stems are permitted to drop to their lowest position in the dash-pots and thus rotate the valves, cutting off the steam from the cylinder.

The point of cut-off, from nothing to full stroke, is regulated by a parabolic Watts governor, which moves the two sliding wedges mentioned above, against which the triggers strike, earlier or later, in the stroke of the piston.

The points of admission, exhaust and cushioning, are readily adjusted from the outside of the cylinder, or steam chest, by means of sleeves and lock-nuts.

All bearings are bushed with hard phosphor-bronze.

Diagram, Fig. 38, illustrates the steam chest, ports and valves in development; different positions of the valves being shown, corresponding to various points of cut-off, or stroke of piston and eccentric.

CLASS A ENGINE. DIAGRAM OF VALVE MOTION.

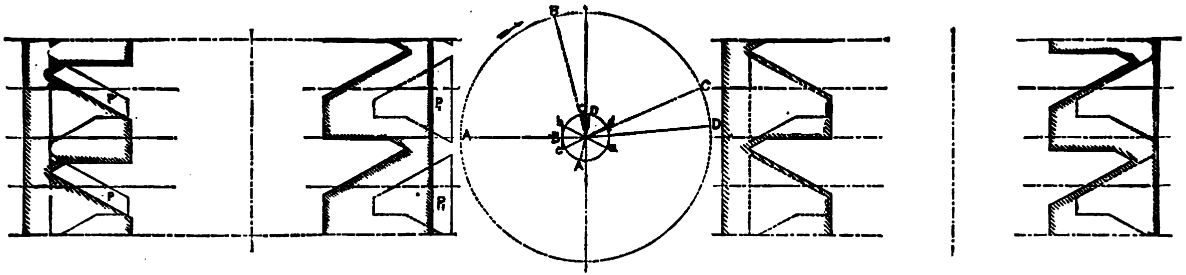


Fig. 38.

Reference to letters.

A B C D, track of crank pin.

a b c d, track of a point on valve, when cutting off at B, or at the same point, relatively, on return stroke.

A, position of crank and eccentric, when admitting steam, no lead.

B, position of crank and eccentric, when cutting off steam.

C, position of crank and eccentric, closure of exhaust port, compression begins.

D, position of crank and eccentric, opening of exhaust port, release begins.

a–d or *b–c*, amount of rotation of valve.

a–c or *b–d*, sliding motion of valve.

P, port open for admission of steam.

P_1, port open for exhausting steam.

Diagram of valve motion, steam chest and valves developed. Page 272.

THE TRENTON EN

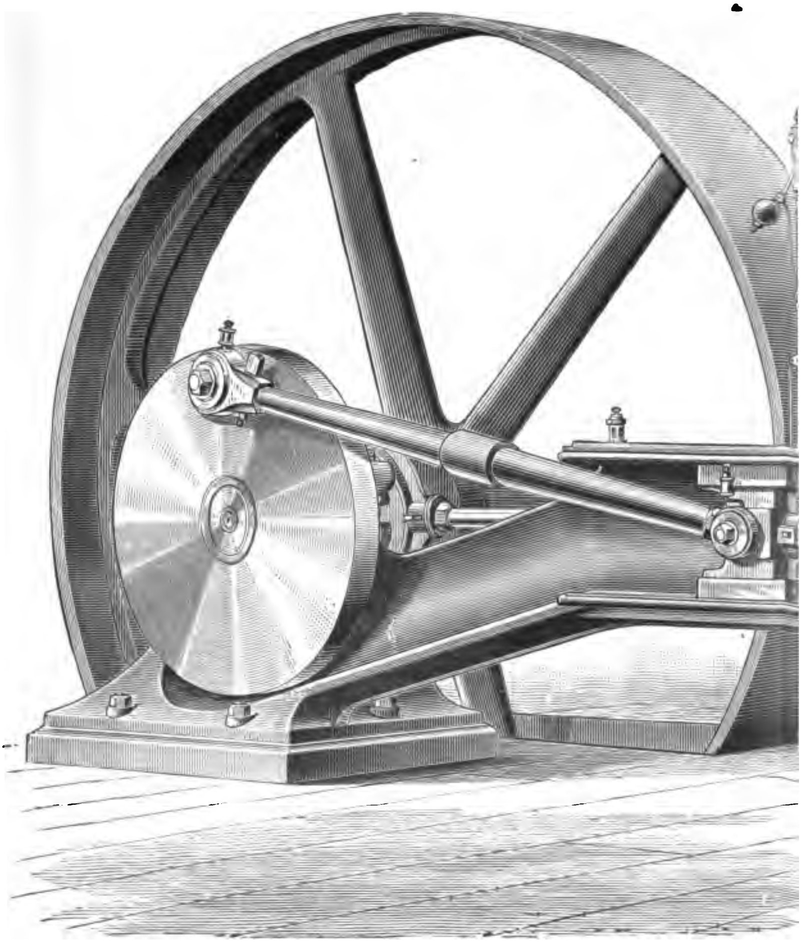
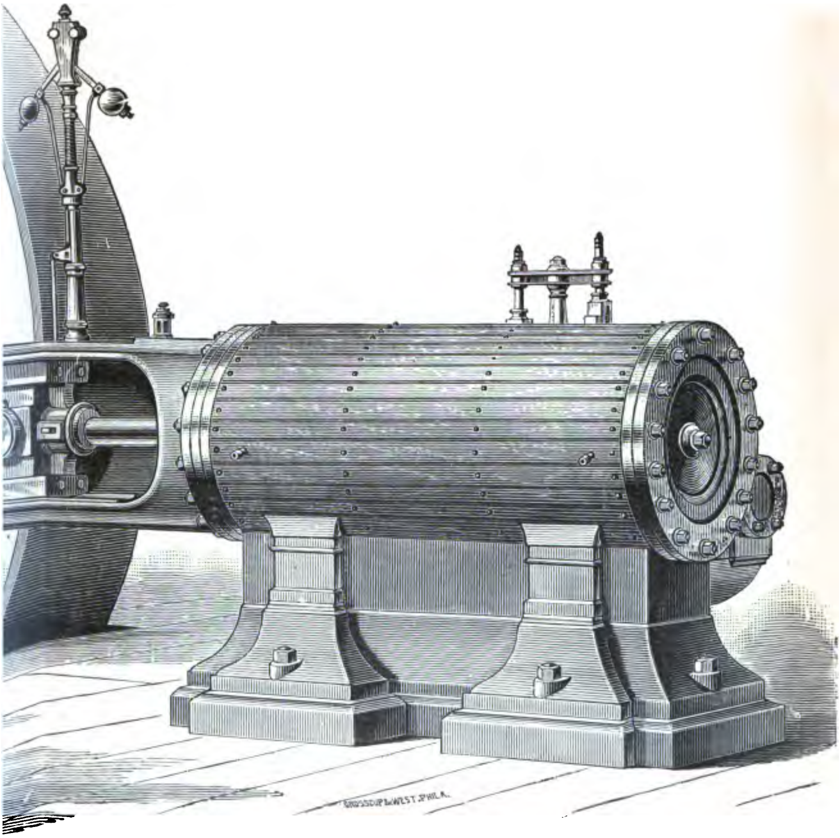


Fig. 39. I

FRONT V

ON ENGINE. CLASS A.


GROSSCUP & WEST. PHILA.

39. Page 272.

ONT VIEW.

THE TRENTO

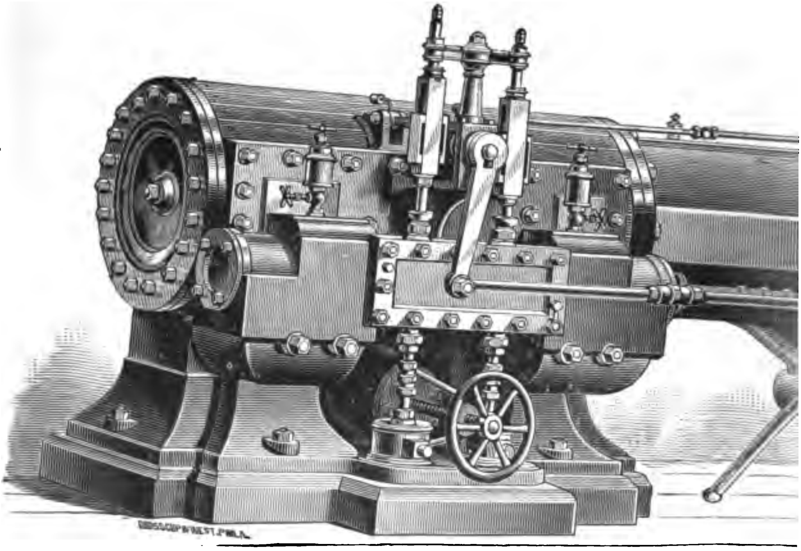


Fig.

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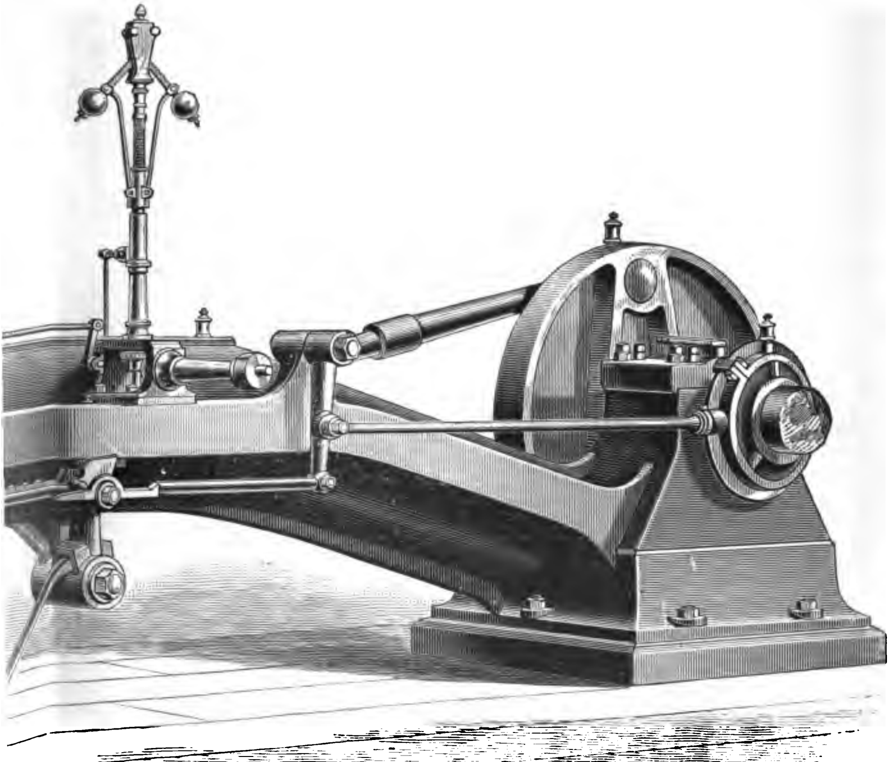
NTON ENGINE. CLASS A.



ig. 40. Page 272.

REAR VIEW.

THE TRENTON EN

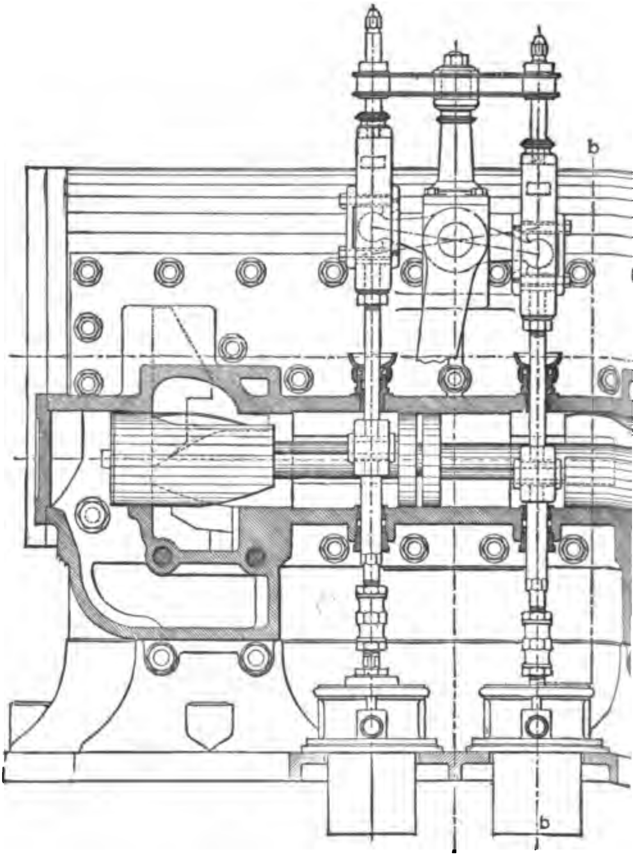
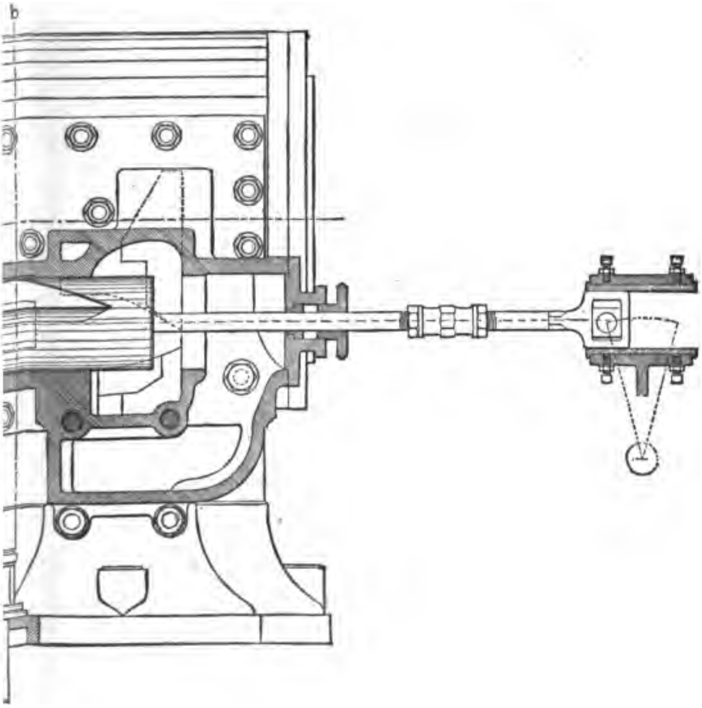


Fig. 41.

VERTICAL SECTION THROUGH STEAM CH

N ENGINE. CLASS A.



41. Page 272.

M CHEST; AND SIDE ELEVATION OF CYLINDER.

CLASS A ENGINE.

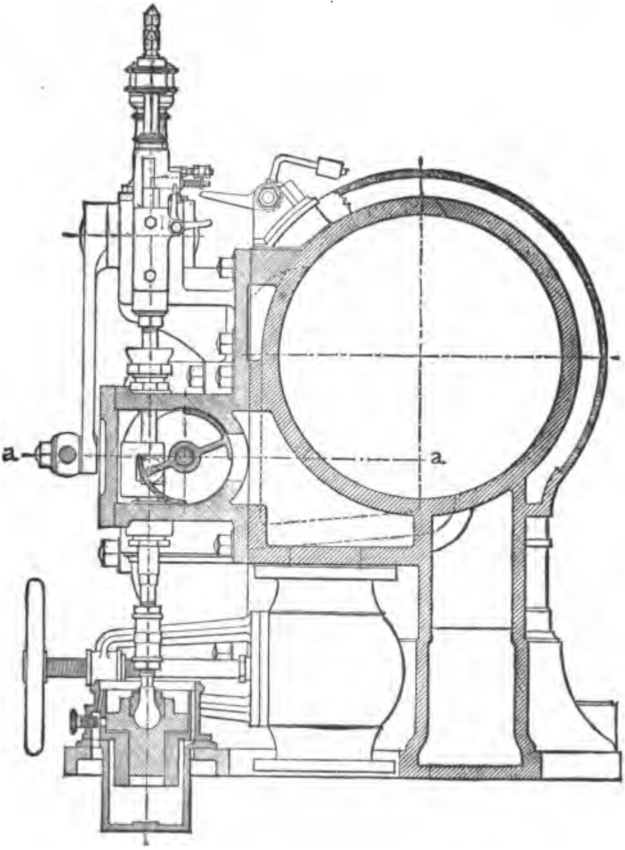
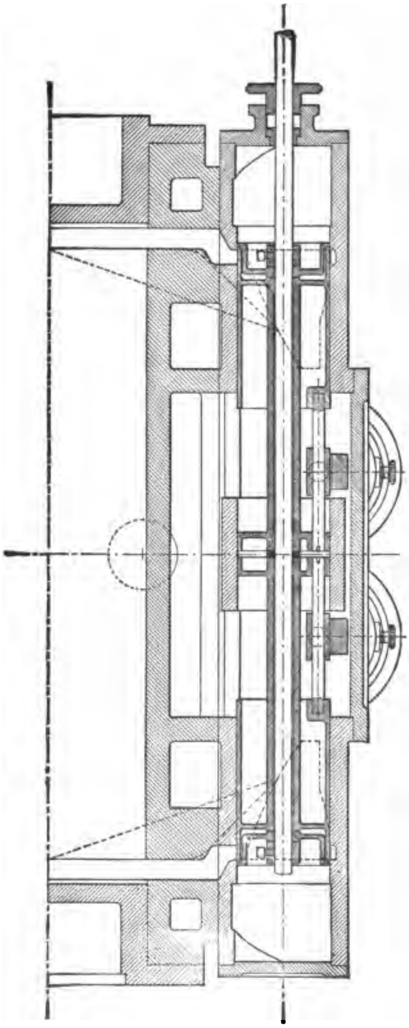


Fig. 42.

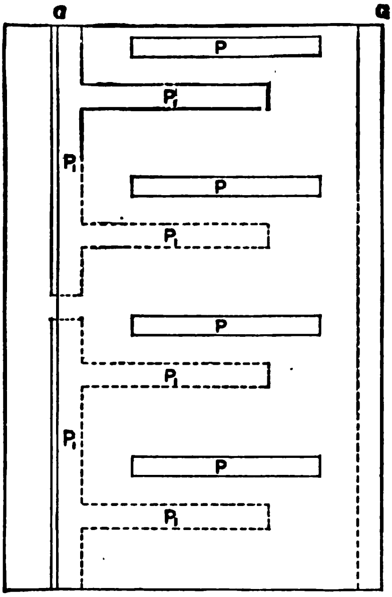
Transverse section through cylinder and steam chest.

CLASS A ENGINE.

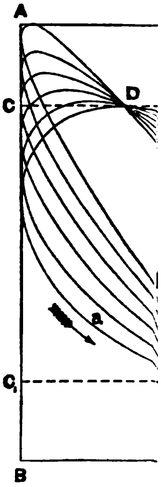


Sectional plan through steam chest and half of cylinder, on line *a-a*.

Fig. 43.



CLASS I



Fig

DIAGRAM OF

Referenc

A A, Direction and length of sliding motion.

A B, Direction and length of rotating motion.

a, Track of valve when cutting off at the beginning of the stroke.

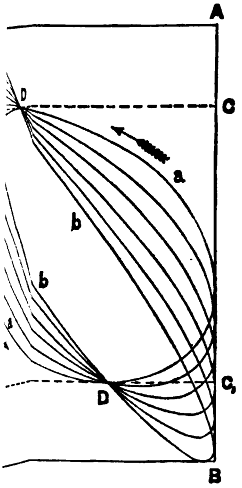
b, Track of valve when cutting off at $\frac{6}{10}$ of the stroke.

Lines between *a* and *b* are various tracks described by a point on the valve, when cutting off at different points of the stroke.

Distances CA or $C_1 B$ show the width of port for the admission of steam, when cutting off at $\frac{6}{10}$ of the stroke.

D, Position of valve at the beginning of the stroke ; or the point of admission.

ASS B ENGINE.



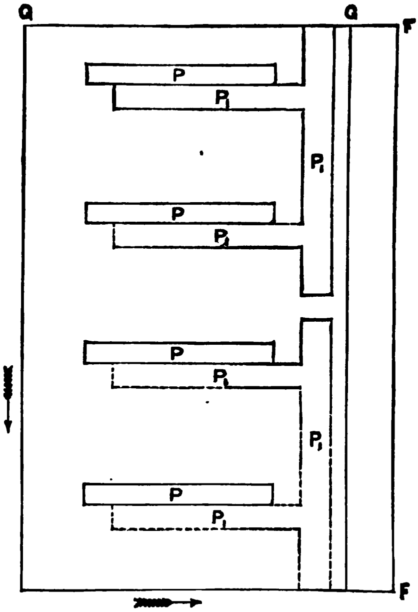


Fig. 44.

M OF VALVE MOTION.

rence to Letters.

stroke.

oint on

mission

e point

P, Ports in valves.

P_1, Ports in steam chest; also steam passages.

F F, Circumference of steam chest bore, or of valves.

G G, Length of valve.

H H, Exhaust openings. } Figs. 47 and 48.

K, Live steam. }

The scale of the diagram, showing the track of valves, is four times that of the steam chest, or of the valves.

CLASS B ENGINE.

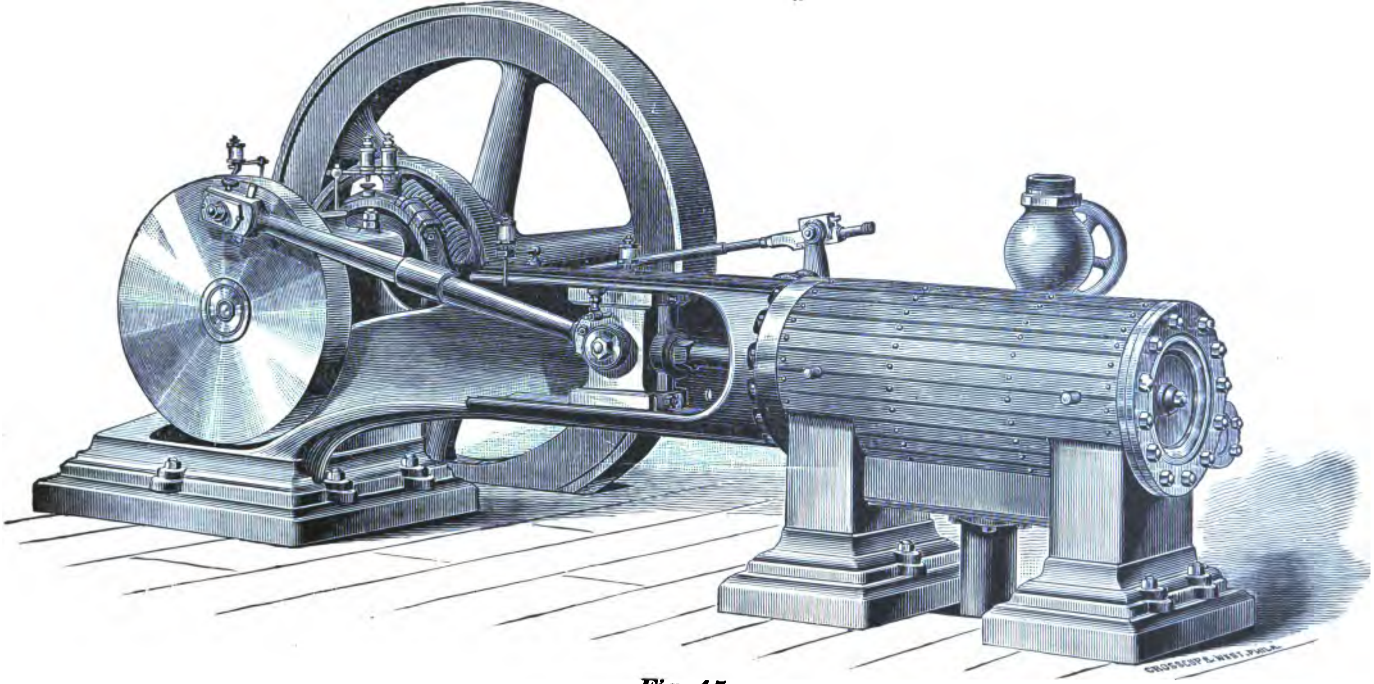

CROSSCUP & WEST, PHILA.

Fig. 45.

FRONT VIEW. Page 275.

CLASS B ENGINE

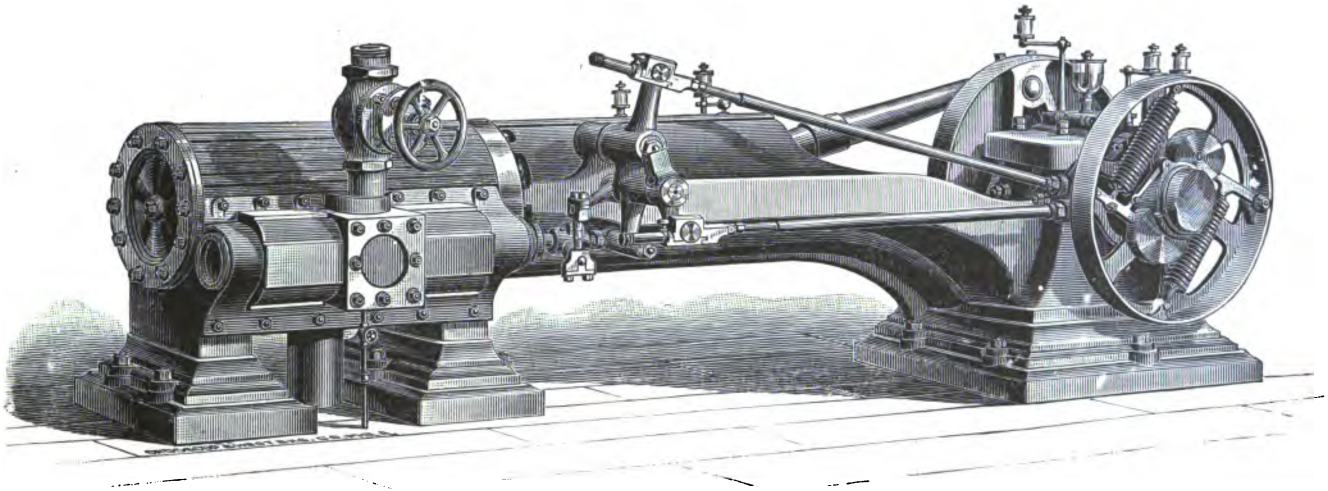


Fig. 46.

REAR VIEW. Page 275.

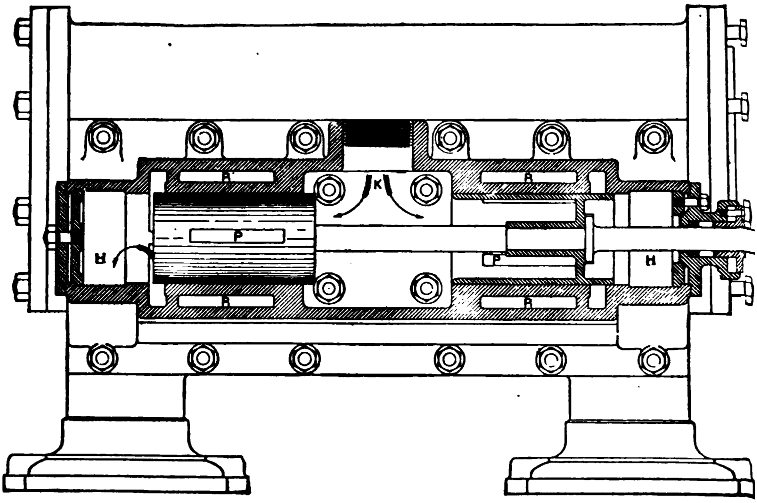
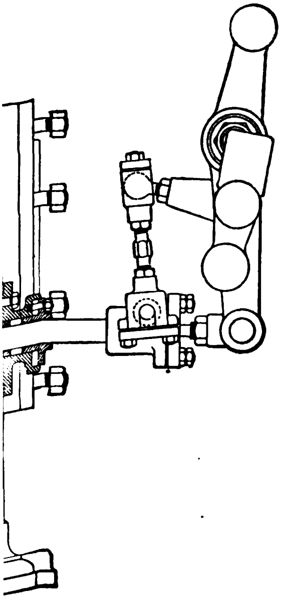


Fig. 47.

VERTICAL SECTION THROUGH
STEAM CHEST.

B ENGINE.



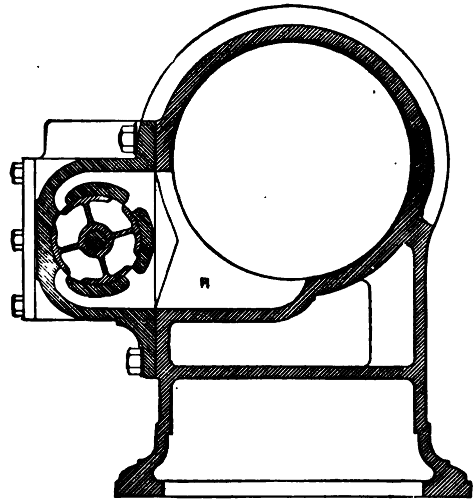


Fig. 48.

TRANSVERSE SECTION THROUGH CYLINDER AND STEAM CHEST.

(Page 275.)

TRENTON ENGINE—CLASS B. AUTOMATIC CUT OFF.

General Description.

Figs. 44 to 48.

This engine is similar in general design to engines of Class A, differing, however, in the valve gear and governor.

The valves, two in number, are similar to those of Class A engines, except that they are fastened to the valve stem, having each four longitudinal ports with steam edges parallel to axis of valve stem instead of helical.

The valves are cylindrical and balanced with their axis parallel to center line of cylinder. They have a simultaneous sliding and rotary motion, the sliding action operating to exhaust and cushion at the outer ends of the valves, and the rotary action the admission and cutting off the steam through the four ports at the same time.

The sliding motion is obtained by means of a positive eccentric on the crank shaft, giving a reciprocating motion to a duplex rock-arm, which carries a secondary bell crank rock-arm operated by an automatic adjustable eccentric attached to the crank shaft governor. This secondary bell crank rock-arm gives the rotary motion to the valves, and by its double motion accelerates the opening of the ports to admit and cut off the steam; so that a full port is obtained at $\frac{1}{80}$ part of the stroke of the piston.

This rapid opening of the ports is one of the most important features of these engines.

All parts of the valve gear are adjustable for wear and are noiseless.

The variation of the point of cut-off is from nothing to two-thirds the stroke.

The governor, which actuates the rotative motion of the valve, is of the "crank-shaft governor" kind, being a disc attached to the crank shaft, and which contains weight, having a centrifugal force from the rotation of the shaft, and held in equilibrium by helical compression springs.

The eccentric, attached to the governor, has a transverse sliding motion across the crank shaft, in such a direction that the point of admission of steam is unvarying, no matter at what part of the stroke it is cut off.

The adjustment between the centrifugal force of the weights, and the resistance of the springs, is so nicely provided for, that a change in speed of three per cent. will cause a change in the position of the weights equal to their extreme range, or a variation of the point of cut-off from nothing to $\frac{2}{3}$ stroke.

This insures regularity of speed under the greatest variations of load.

This engine must present itself most favorably,

CLASS C ENGINE.

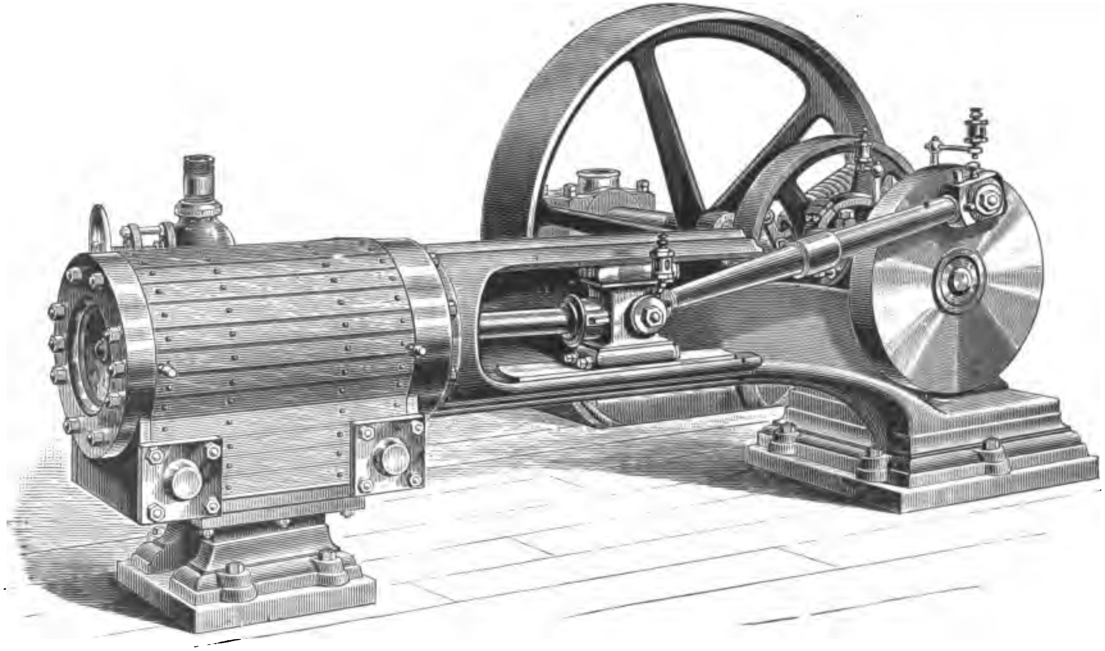


Fig. 49.

FRONT VIEW. Page 277.

CLASS C ENGINE.

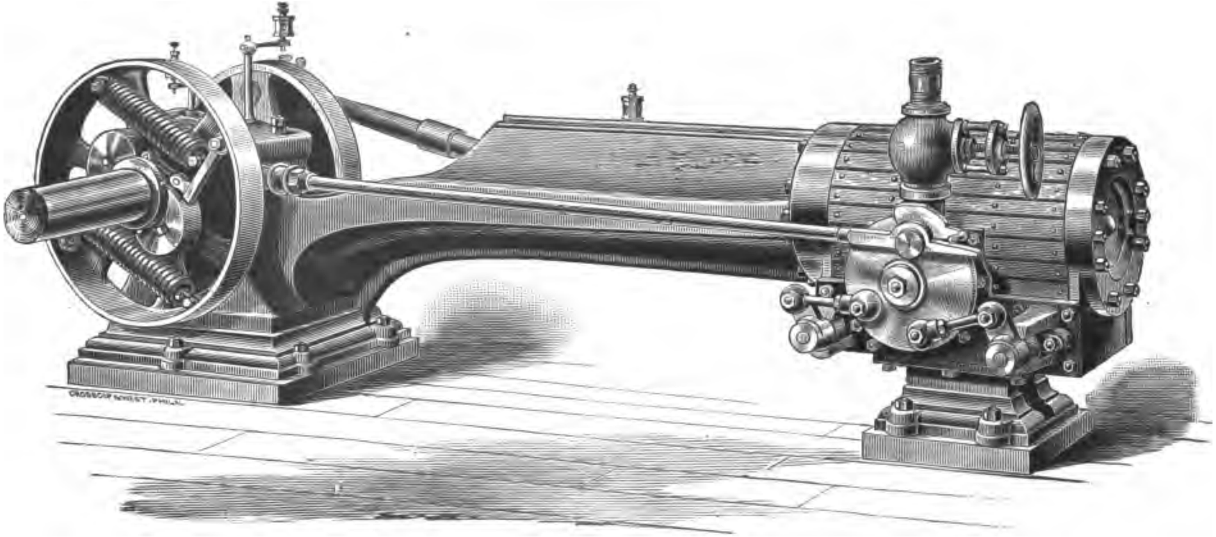

CROSSCUP & WEST, PHILA.

Fig. 50.

REAR VIEW. Page 277.

as compared with engines of other builds, because of its great simplicity in the valve gear, and general excellence of design and construction.

TRENTON ENGINE—CLASS C. AUTOMATIC CUT-OFF.

General Description.

Figs. 49, 50, 51.

The intention of the builders to have the general design of the different classes of engines similar, has been extended to Class C; the bed or frame corresponding to those already described.

The valve gear differs from those of the other classes as follows:

The valves are two in number and are placed transversely across the bottom of the cylinder. They are of the Corliss or rotary type, improved by patent double ports.

They are rotated by an eccentric and wrist-plate, which opens the valves for the admission of steam to the cylinder very quickly, and retards their closing during the exhaust; both important factors in the economical distribution of steam in a cylinder.

We direct especial attention to the valves as illustrated by Fig. 51.

It will be seen that there is obtained double the amount of port opening in the earlier parts of the stroke, than is found in any other valve of this character.

This insures the highest initial steam pressure in the cylinder, and tends towards the greatest economy of steam, and consequently of fuel.

The speed and power of this class are regulated by a centrifugal governor on the crank shaft, similar to that used for Class B engines.

These engines are designed for high speed with a varying intensity of load.

CLASS C ENGINE.

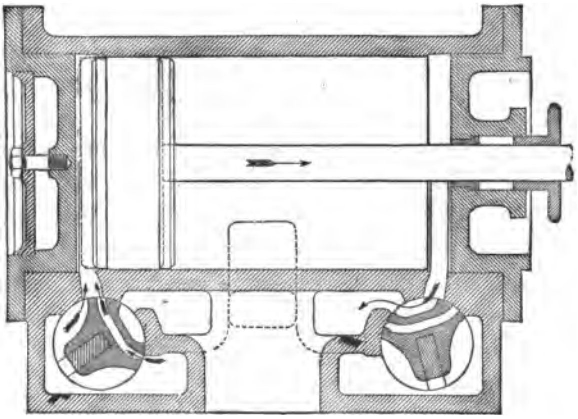


Fig. 51.

VERTICAL LONGITUDINAL SECTION THROUGH CYLINDER AND VALVES. FULL PORT AT $\frac{1}{17}$ OF STROKE.

CLASS D ENGINE.

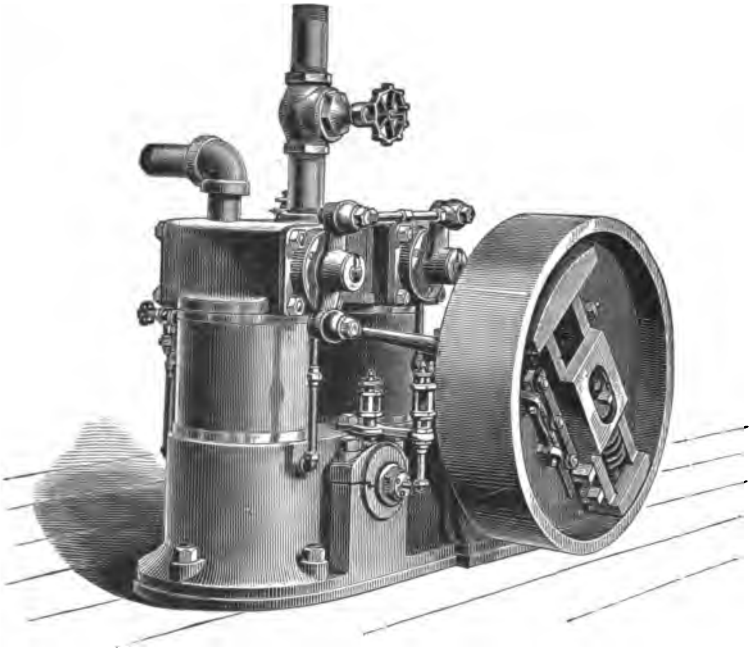


Fig. 52.

FRONT VIEW. Page 279.

CLASS D ENGINE.

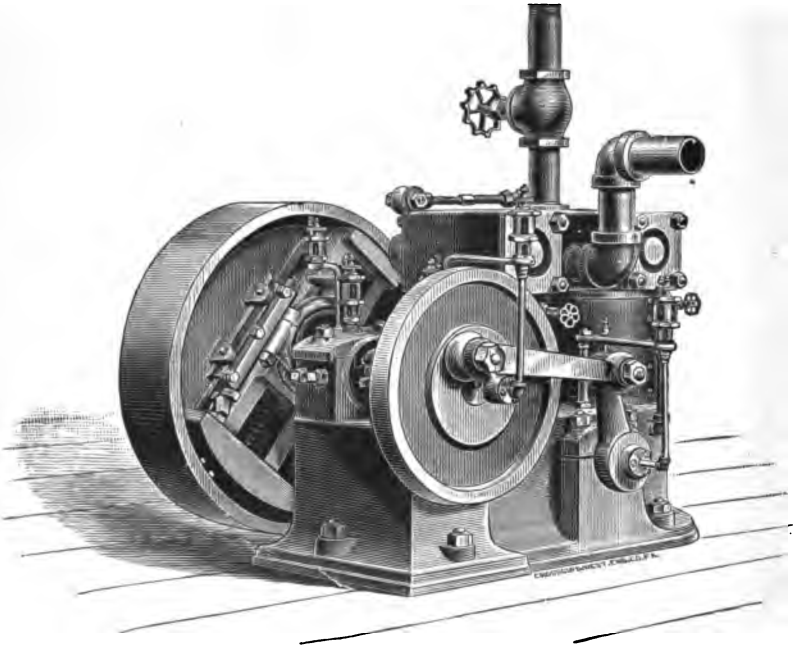


Fig. 53.

REAR VIEW. Page 279.

TRENTON ENGINE—CLASS D. AUTOMATIC CUT-OFF.

General Description.

Figs. 52 and 53.

Class D engines differ materially from the others. The illustrations fully explain the design. The bed plate is self-contained, requires no extra foundation for bearings. Motion is imparted to the crank shaft from two cylinders, each of which is single acting, admitting steam at the top only. The pistons are attached to each end of a walking-beam, by means of piston rods having spherical ends. These work in Babbitt-metal bearings, and adjust themselves to any position they may assume; the ball and socket joints thus obtained tend to retain the lubricants and prevent cutting.

The valves are double-ported rocking valves, similar to the well-known Corliss valves, but having the auxiliary port; the speed of the engine is controlled by a governor similar to those already mentioned.

All parts can be oiled whilst running.

SIZES, SPEEDS AND HORSE-POWERS.

Classes A, B, C and D.

The table on page 281 gives the necessary data, from which to choose the size and class of engine.

The speeds may be varied to suit special cases. Those given, and which we recommend, are consistent with the greatest economy in the use of steam.

Weights and dimensions of band or fly wheels are not stated, because of their great diversity, due to different requirements.

The column headed "rated capacity" gives the actual effective horse-power (H. P.) these engines will develop, with the ratio of expansion or point of cut-off within economical limits. As will be seen, it is considerably less than the maximum power.

The horse-powers given are based upon a boiler pressure of 70 lbs. per square inch, and a modification of the theoretical mean effective pressure (M. E. P.) of the steam in the cylinder throughout the stroke, due to diminished initial pressure, back pressure, and cushioning, as shown in the annexed table.

Hence the horse-powers given represent the actual power the user may depend upon obtaining from these engines.

ACTUAL AVERAGE PRESSURES OF STEAM IN CYLINDER.
BOILER PRESSURE, 70 LBS.

Point of Stroke at which Steam is cut off.	Theoretical M. E. P.	Percentage of M. E. P. adopted.		Resultant average pressure to determine actual power.	
		Classes A and B.	Classes C and D.	Classes A and B.	Classes C and D.
$\frac{2}{10}$	36.43 lbs.	85	55	30.96 lbs.	20.00 lbs.
$\frac{3}{10}$	46.22 "	85	70	39.28 "	32.35 "
$\frac{6}{10}$	63.25 "	85	75	53.76 "	47.44 "

Class.	Dimensions of Cylinder in Inches.		Revolutions per minute.	Piston Speed in Feet per Minute.	Area of Piston in Square Inches.	Horse power Actually Developed when Steam is Cut off at			Diameter of Fly wheel.	Weight of Fly wheel.	Approx. Shipping Weight.	Rated Capacity.
	Bore.	Stroke.				$\frac{2}{10}$ Stroke.	$\frac{3}{10}$ Stroke.	$\frac{5}{10}$ Stroke.				
A.	32	60	60	600	802.24	452.79	564.23	788.15	24′ 0″	42,000	105,316	**600**
	30	60	60	600	706.26	397.62	504.27	692.13	24′ 0″	38,500	96,029	**550**
	28	48	75	600	615.75	346.67	439.64	603.43	18′ 0″	33,250	74,792	**475**
	26	48	75	600	530.93	298.91	379.08	520.31	18′ 0″	28,000	65,555	**400**
	24	48	75	600	452.39	254.69	323.00	443.34	16′ 0″	24,500	58,431	**350**
B.	22	42	85.7	600	380.13	214.01	271.41	372.53	16′ 0″	18,000	46,956	**300**
	20	42	85.7	600	314.16	176.87	224.31	307.87	16′ 0″	15,000	40,920	**250**
	18	42	85.7	600	254.46	143.26	181.68	249.37	16′ 0″	12,000	35,188	**200**
	15	24	150	600	176.71	99.48	126.17	173.17	8′ 0″	9,000	18,956	**150**
	14	24	150	600	153.94	86.64	109.86	150.46	8′ 0″	7,000	17,300	**125**
	13	24	150	600	132.73	76.56	94.80	129.72	8′ 0″	6,000	15,400	**100**
	12	20	180	600	113.09	63.67	80.74	110.84	7′ 0″	4,800	11,136	**80**
	11	20	180	600	95.03	53.49	67.86	92.88	7′ 0″	4,200	10,500	**70**
	11	16	200	533⅓	95.03	47.54	60.31	82.29	6′ 0″	3,600	7,600	**60**
	10	16	200	533⅓	78.54	39.25	49.86	68.21	6′ 0″	3,000	7,061	**50**
	10	12	225	450	78.54	33.12	42.07	56.55	5′ 0″	2,000	5,000	**40**
	9	12	225	450	63.62	26.82	34.06	46.62	4′ 3″	1,500	4,000	**30**
C.	11	16	187.5	500	95.03	28.79	46.57	68.30	6′ 0″	2,250	6,500	**45**
	10	12	225	450	78.54	21.42	34.75	50.80	6′ 0″	1,750	4,500	**35**
	9	12	225	450	63.62	17.32	28.17	41.17	4′ 3″	1,250	3,750	**25**
	8	10	270	450	50.26	13.68	22.14	32.49	4′ 0″	1,000	3,500	**20**
D.	6	6	425	425	28.27	7.27	11.77	17.25	2′ 0″	300	1,400	**12**
	5	5	450	375	19.63	4.46	7.20	10.57	2′ 0″	200	1,000	**8**
	4	4	475	316⅔	12.56	2.31	3.89	5.70	2′ 0″	150	750	**4**
	3	3	500	250	7.07	1.07	1.74	2.54	2′ 0″	100	500	**2**

CHAPTER XV.

THE GREENE AUTOMATIC CUT-OFF ENGINE.

FIGS 54–56 show the improved Greene Automatic cut-off engine, of which Providence (R. I.) Steam Engine Company are the sole builders.

The bed-plate is of the girder pattern, symmetrical in appearance, and of ample strength.

The slides are cast *separate* and secured to bed-plate by dowels and bolts. The main journal boxes are made in four pieces, and provided with set screws and check nuts which permit of convenient and accurate adjustment. The governor is of the Porter pattern, and is driven by a flat belt from the main shaft.

The valve-gear is detachable, and is so controlled by the governor that the cutting off may be effected from zero to three-quarters of the entire stroke.

The valves are four in number—two steam and two exhaust—and are of the *flat-side pattern.*

The power which moves them is applied parallel to and in line with their seats, so that they cannot rock or twist, thus obviating the tendency to wear unevenly.

The steam-valves when tripped are shut by a combined action of a weight and the pressure of

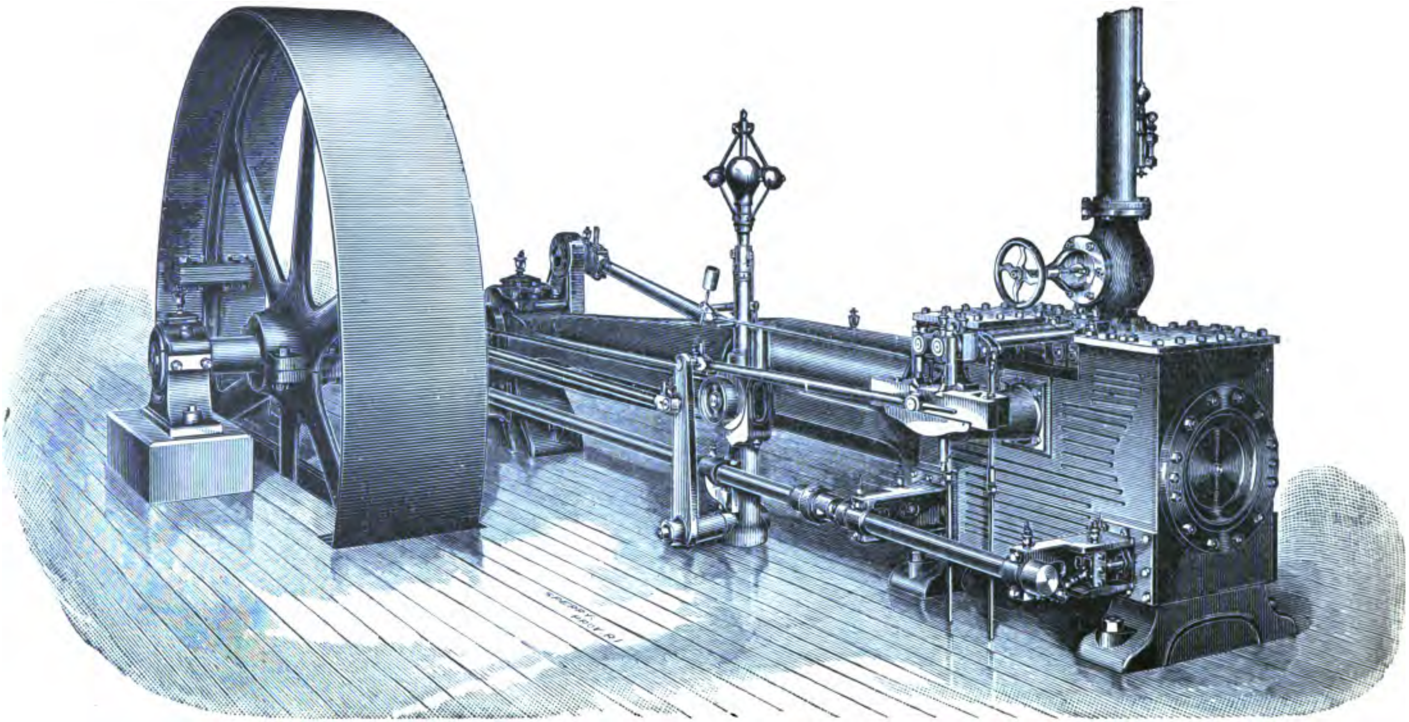


Fig. 54. Page 283.

IMPROVED GREENE AUTOMATIC CUT-OFF ENGINE.

the steam on the large valve-stems, thereby insuring a quick cut-off, and the positive closing of the port, under all circumstances of speed and pressure.

The steam-valves are operated by toes, on the inner ends of two rock-shafts that connect with the valve-stems outside the steam-chest. The outer ends of the rock-shafts are furnished with steel-tipped toes.

There is a sliding bar carrying tappets, which receives a reciprocating rectilinear motion from an eccentric on the main shaft.

Below the sliding bar is a gauge-plate connected with the governor, which receives an up and down motion from a reverse action of the governor balls.

The tappets in the sliding bar are attached to the gauge-plate, and elevated or depressed in the bar by the action of the governor.

As the sliding bar moves in the direction of the arrow, one of the tappets is brought in contact with the inner face of the toe on the rock lever, causing it to turn on its axis, thereby opening the steam-valve at one end of the cylinder. At the same moment the other tappet comes in contact with the outer face of the other toe, and as the surfaces are beveled, the toe is forced up into the cocket until the tappet passes under, when it drops by gravity alone into its original position, to be operated upon in its turn, when the motion of the sliding bar is reversed.

As a result of this motion the tappets always give the valves the same lead, and as the bar moves in a straight line, while the toe describes the arc of a circle, the tappet will pass by and liberate the toe, which is brought back to its original position by a weight, and the steam pressure on the large valve-stem, which thus closes the valve and cuts off the steam. The liberation of the toes will take place sooner or later, according to the elevation of the tappets: that is, the lower the tappets are, the sooner the toes will be liberated, and *vice versa.* By the elevation or depression of the gauge-plate, the period of closing the valves is changed, while the period of opening them remains the same. The adjustment of the gauge-plate is effected directly by the governor.

Both the exhaust-valves and seats are convenient of access, and removable from the outside of cylinder. The valves receive their motion from a seperate eccentric, thus allowing of easy adjustment, without interference with the steam-valve mechanism. All the connections are on the outside, are few in number, and have ample bearing surfaces, insuring freedom from rapid wear and derangement.

A safety stop-motion is combined with the governor, preventing the admission of steam, should the governor belt run off or break. The cross-head gibs are directly opposite the centre of pin, thus avoiding any cross strain upon the piston rod; a lack of attention to this point has been the cause of many serious accidents.

THE GREENE ENGINE.

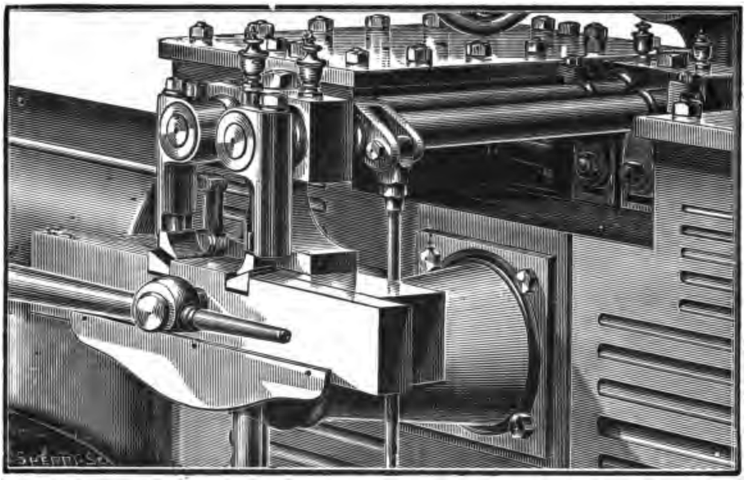


Fig. 55.
SHOWS THE STEAM VALVE GEAR IN PERSPECTIVE.

The steam-ports are large, thus insuring the full pressure of steam to the point of cut-off.

The engine is extremely sensitive to the action of the governor, and is, therefore, particularly adapted to those situations where perfect regulation is required.

All parts are well proportioned, made of the best material, accurately fitted, and highly finished.

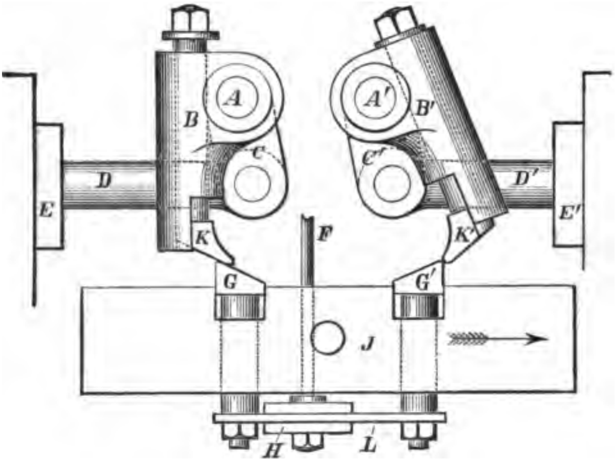


Fig. 56.

SHOWING ACTION OF STEAM-VALVE GEAR.

A A' are the two rock levers; B B' the sockets attached to them and within which the toes K K' are contained; C C' the arm of the levers attached to valve-stems; D D' the valve-stems: E E' the steam-chest; F the governor rod; GG' the tappets; L the gauge-plate; H gauge-plate guide, and J the sliding-bar.

The sliding bar J is shown moving in the direction of the arrow, causing the tappet G' to operate upon the toe K', giving an opening movement to the valve-stem D', the tappet G meanwhile passing under and lifting the toe *K*. The amount of opening of steam-valves is determined by the height of the tappets *G G'*, the positions of which are always under control of the governor. The toes drop by gravity, no springs being used, thus relieving the governor of all strain while tripping.

CHAPTER XVI

AGRICULTURAL ENGINES.

EVERY large farm or plantation should be supplied with its own Portable Steam Engine, from 5 to 15 horse power, depending upon amount to do.

For threshing, ginning, grinding, sawing wood, and, indeed, anywhere that stationary horse-power is much used, the engine may be employed to good advantage, as it will do more work for the same cost than horse-power.

Portable Engines are among the best made in America, and while embracing all the latest improvements, they are simple in construction, economical in fuel, easily kept in order, and very durable. Charcoal pig-iron is used in the castings of engines, the connections are made of steel and bronze, the pistons double-ringed and self-packing, and the boilers are made from charcoal iron or steel, with tensile strain of not less than 50,000 pounds; fire-boxes always of best steel.

In purchasing a steam engine, it is of the first importance to get the best, made by a responsible manufacturer who values his reputation too highly to use inferior work or material, as, aside from the endless trouble and loss entailed by defects, there is danger to life and limb.

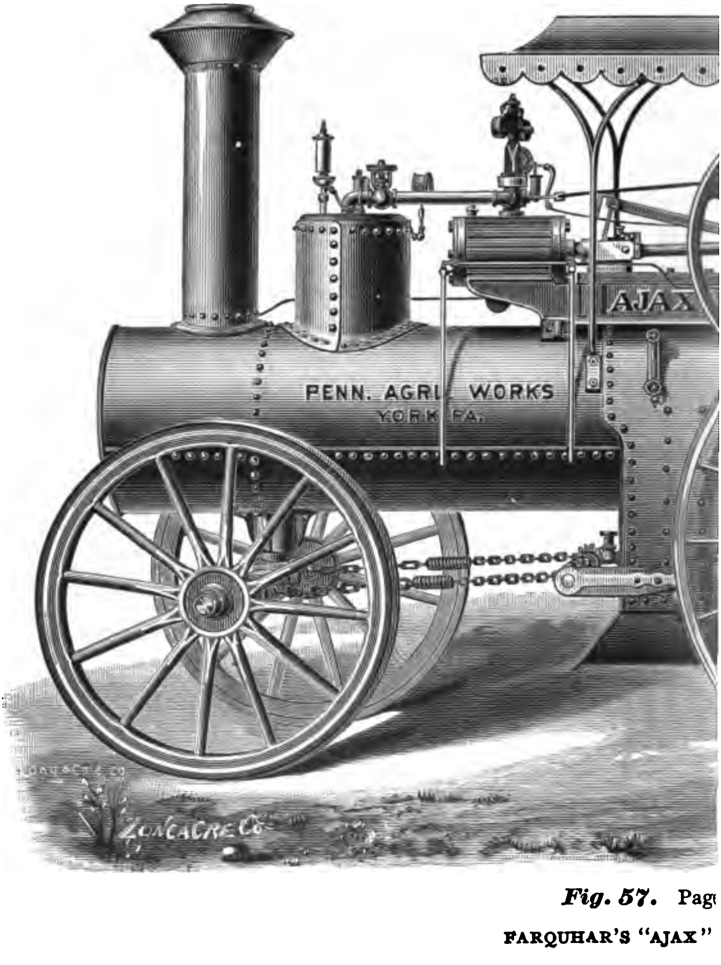
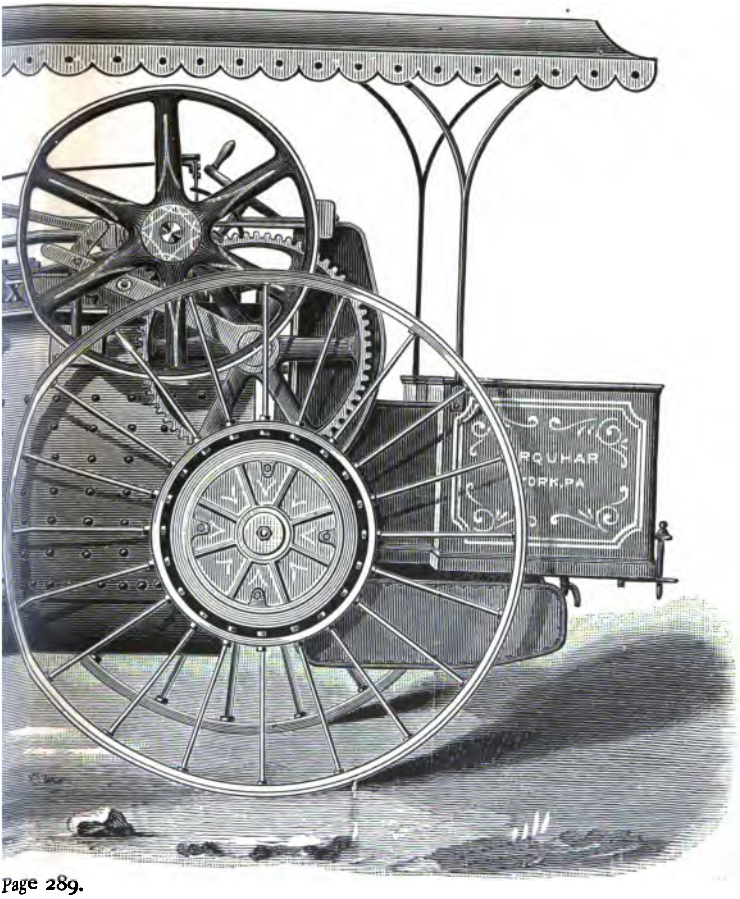


Fig. 57. Pag

FARQUHAR'S "AJAX"



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X" TRACTION ENGINE.

A good steam engine is much more economical than animal power when in use, and, eating no oats, costs nothing when idle. It will pay on any well-conducted farm of 100 acres or over.

Steam power costs so much less than animal power, even on the farm, that it is only necessary to give to the public a good engine that is safe, simple, and not too expensive, for the steam engine to become as universal as the reaper, cotton-gin, or thresher. Such an engine is as sure eventually to take the place of animal power on the farm as the locomotive has of the stage-horse.

Farmers cannot afford to keep stock to do their work when steam power can be had at these prices. A six-horse steam engine with vertical boiler costs $350, while six good horses are worth, say, about $800, and $200 to $300 more annually for feed and attention. The engine puts you to no expense when not in use, and is in no danger from thieves; horses must be cared for always, and will eat nearly as much when idle as when at work.

FARQUHAR'S "AJAX" TRACTION ENGINE.

(Fig. 57.)

The great demand for traction engines has induced many builders to undertake their manufacture who have had no experience in this line, or in fact very little in the more simple portable

PRINCIPAL DIMENSIONS FOR PORTABLE ENGINES, MOUNTED AND ON SILLS.

	CYLINDER.	BOILERS.						STACK.		FLY-WHEEL.		SMALL PULLEY.			ON SILLS.
Horse Power.	Bore and length of stroke.	Diameter.	Length of fire-box in inches.	Width of fire-box in inches.	Height of fire-box in inches.	Number and size of tubes.	Length of tubes in inches.	Diameter of stack in inches.	Length of stack in feet.	Diameter in inches.	Width of face in inches.	Diameter in inches.	Width of face in inches.	Revolutions per minute.	Estimated weight.
3 to 4	4½×6	24	29	20	25	22 2 in.	44	8	8½	24	4	16	6	225	2,500
4 to 5	5 ×6	24	29	20	25	22 2 in.	50	8	8½	33	5	16	6	215	2,800
5 to 6	5½×8	26	29	22	27	24 2 in.	52	8	8½	33	5½	16	6	200	3,100
6 to 8	6 ×8	26	31	22	28	26 2 in.	55	9¾	8½	36	6	16	8	200	3,400
8 to 10	7 ×10	26	38	22	28	32 2 in.	63	9¾	8½	40	7	18	8	190	3,800
10 to 12	8 ×10	30	41	25	29	36 2 in.	68	9¾	10	44	8	18	8	180	5,050
12 to 15	9 ×11	33	41	28	30	26 3 in.	76	12	12	48	10	30	8	175	6,500
15 to 18	9 ×15	36	49	31	32	28 3 in.	93	15	14	60 or 72	12	30	8	150	7,500
18 to 23	9½×15	36	49	31	32	30 3 in.	96	15	14	60 or 72	12	to order.	to order.	150	8,200
20 to 25	10 ×15	39	52	34	39	32 3 in.	103	15	16	72	13	to order.	to order.	140	9,500
25 to 35	10 ×18	39	52	34	39	36 3 in.	117	19	18	72	14	to order.	to order.	130	11,000
30 to 40	11 ×20	42	58	36	47	46 3 in.	110	19	18	84	15	to order.	to order.	110	12,000

engines; and the result is, a great variety of so-called traction engines have been put on the market which are poor specimens of mechanism, and must bring the purchaser only disappointment and expense.

It is absurd to stick a few cog-wheels on an ordinary portable engine and call it a traction engine. A good traction engine is not only self-propelling, but must be able to draw at least its own weight over almost any kind of road, and on a very good road three or four times its own weight would be a fair load. To accomplish this requires a road-locomotive in the fullest sense of the word, and while there are engines that will do it, the great majority of them will not, and some will not pull an ordinary separator over a good hill.

It is very poor economy to buy such an engine for the sake of saving a few dollars. The lowest priced machine is rarely the cheapest.

The Farquhar "Ajax" Traction Engine has been brought to a state of perfection attained by few others. A careful examination of the illustration will show the care with which this engine has been designed. All handles, levers, valves, etc., are within easy reach of the engineer.

The boiler is supplied with an extension smoke-box, within which is placed a regular locomotive spark-catcher, which will prevent escape of sparks,—a very important thing.

Spark-catchers placed in the top of the stack are good for nothing, as they soon become so clogged with dirt that the engineer is compelled to remove them to obtain sufficient draft, thus leaving the engine without a spark-arrester.

The boiler, which is the most important part of any steam machinery, is very carefully made.

The fire-boxes are of steel plate made especially for the purpose.

THE BOILER.

They make their traction engine boilers from the best open-hearth steel plates (the Pennsylvania Railroad Co,, use the same brand of steel in their locomotives). By the use of steel the boiler will stand a higher steam pressure than would be safe with the heaviest refined iron plates. These boilers are made with the utmost care by experienced workmen, carefully tested.

The crown sheet is powerfully braced and is made with an incline of several inches toward the fire-door end; this makes it possible to descend steep hills without the crown sheet becoming bare. The water glasses are placed at the highest point, and if water shows at all in the glass the entire sheet is certain to be well covered. The extension smoke box in which the spark-catcher is placed is water-lined; by this improvement these gain several feet of heating surface, besides preventing the plates from burning out.

The fire boxes are large and long, with large doors making them good wood burners and handy to fire. A coil of cross tubes is placed in top of fire box which adds greatly to the steaming capacity.

THE ENGINE.

Is of the most improved center crank pattern. The frame or bed-plates is very heavy and well braced, to insure strength and stiffness. The pedestals are cast solid to the frame, and are made with unusually long bearings; the pedestal boxes are lined with genuine babbit metal. The cross head is of an improved pattern, made with wide bearing surfaces, which are protected from dust and dirt by being sunk into a recess in the bed plate; this recess also serves to hold the oil around the cross-head slide. The connecting rod is hammered iron, forged solid at the cross-head end; the crank end has a strap firmly bolted on with two steel bolts; provision is made at both ends for taking up wear of boxes, which are made of phosphor bronze; these connecting rod boxes are extra long, those for instance on the 7x10 engine being 2½ inches at cross-head end and 3 inches long at crank end.

The cylinders are made large, as they think it best to have cylinders large enough to do the work without carrying such a high pressure of steam.

They set the safety valves to blow off at 115 lbs., and we consider any engine that requires over that pressure to be very badly constructed and undesirable. Cylinders are made of charcoal iron as hard as can be worked; the joints are all ground; no gum is used, gum packing being always liable to blow out.

The crank shaft is made of steel.

The piston head has self-packing rings, which will run for years without leaking, and, when worn out, they can be replaced at small cost and put in by any one; the piston rod and valve rod are made of steel of the best quality; in fact, all parts of the engine are the best that can possibly be made, as there is nothing in the machinery line that gets harder usage or requires better work than an engine used for the road.

Correspondents sometimes ask, "How fast will such an engine travel? How much will it pull over steep hills? How much fuel will it require per day," etc., etc. These questions cannot be answered very definitely unless all the conditions are known.

On a good road six miles per hour can be run, but it is very seldom that any one should go that fast; unless the road is wide and free from turns the engineers will find it very difficult to guide properly at that speed, and if care is not taken would get into trouble. Traction engines should not be designed for great speed; they are not intended for running races: whenever you gain speed you lose power. The "Ajax" is calculated to travel over ordinary country roads with a load at about the speed a good team of horses will go.

The amount that they will pull depends *entirely* on the condition of the road. No traction engine will pull much in deep mud or sand. On a hard road they will pull another engine of about their own weight over any ordinary hill. The engine is powerful enough to slip the drivers on any kind of road. The amount of fuel used depends on the kind of fuel and nature of the work done. The No. 2 engine, for instance, will consume on a day's ordinary threshing, where conditions are reasonably favorable, about 350 to 400 lbs. of bituminous coal; if the engine is forced to its greatest capacity the coal consumed will be more. The amount of water for this size engine used in the same length of time will be from 10 to 12 barrels.

We consider the No. 2, 6½x9, or the No 3, 7x10, the most satisfactory sizes for general purposes; the No. 1 engine is rather light for general use and fast threshing, but is heavy enough to successfully drive a 30-inch separator, while the No. 4 is rather large for threshing, but just the right size for saw mills.

GENERAL DIMENSIONS OF THE "AJAX" TRACTION ENGINES.

		CYLINDER		BOILER.							SMOKE STACK.		FLY WHEEL.			TRACTION WHEEL.	
					FIRE BOX.			TUBES.									
No. of Engine.	Horse-power.	Diameter, inches.	Stroke, inches.	Waist diam.	Length, inches.	Width, inches.	Height, inches.	Length, inches.	Number.	Diameter, inches.	Diameter, inches.	Length, feet.	Diameter, inches.	Width of face, inches.	Revolutions.	Diameter, inches.	Width of face, inches.
1	7-9	6	8	26	31	22	27	55	32	2	10	4	32	6	300	64	9
2	9-11	6½	9	28	34	23	29	58	37	2	10	4	34	7½	275	68	10
3	11-13	7	10	28	34	23	29	68	37	2	10	4	36	8	250	68	10
4	13-15	8	10	30	40	24	29	72	42	2	11	4	40	9	225	72	12

FARQUHAR'S AUTOMATIC ENGINES. (Figs. 58 and 59.)

These engines are the invention of the superintendent of the engine department of these works, who has had a long and varied experience in engine building, and whose reputation as a reliable mechanic and engineer has never been questioned. In designing these engines the greatest care has been taken to insure the most satisfactory results, every known improvement and attachment that good mechanical training together with years of valuable experience could suggest has been added, making an engine that for good workmanship, fine finish, strength and durability, simplicity and handsome appearance, is seldom equaled and never excelled.

The power of these engines depends on the

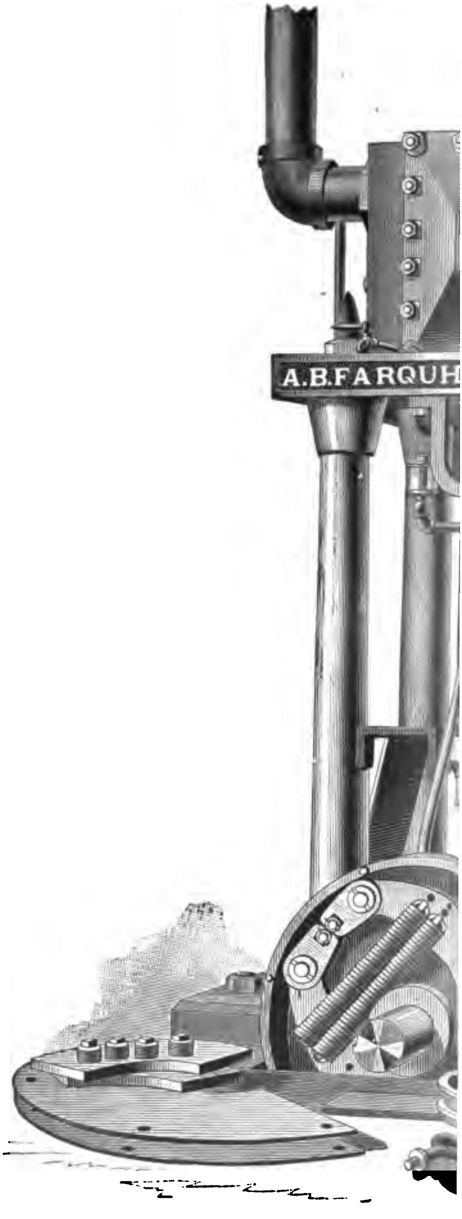
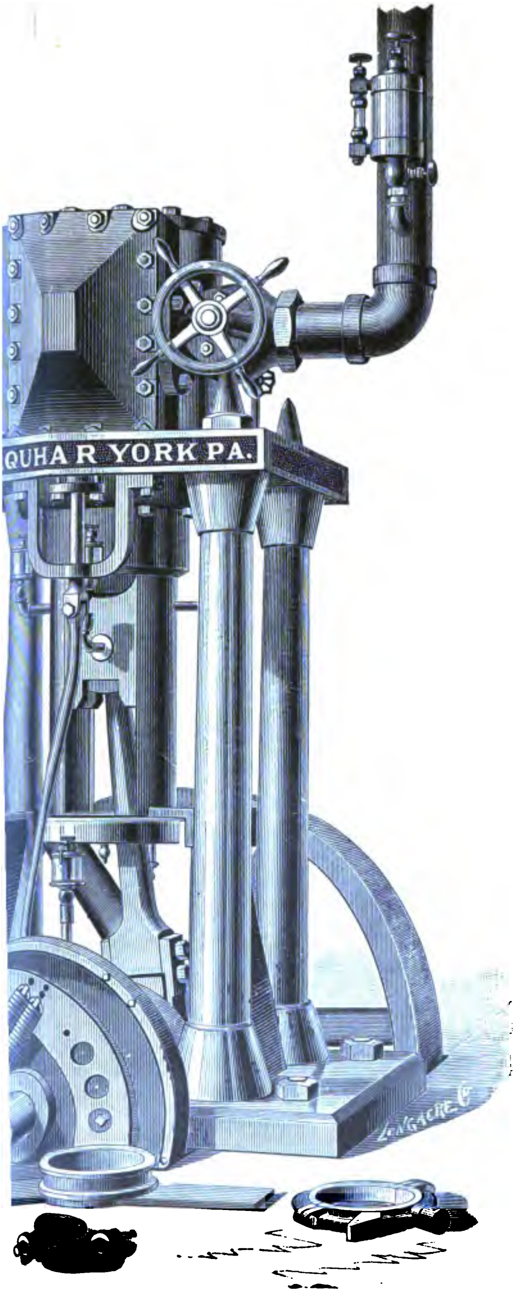


Fig. 58.

A. B. FARQUHAR'S (YORK, PA.), VERTICAL AUT



58. Page 296.

AUTOMATIC ENGINE WITH CRANK SHAFT BELOW.

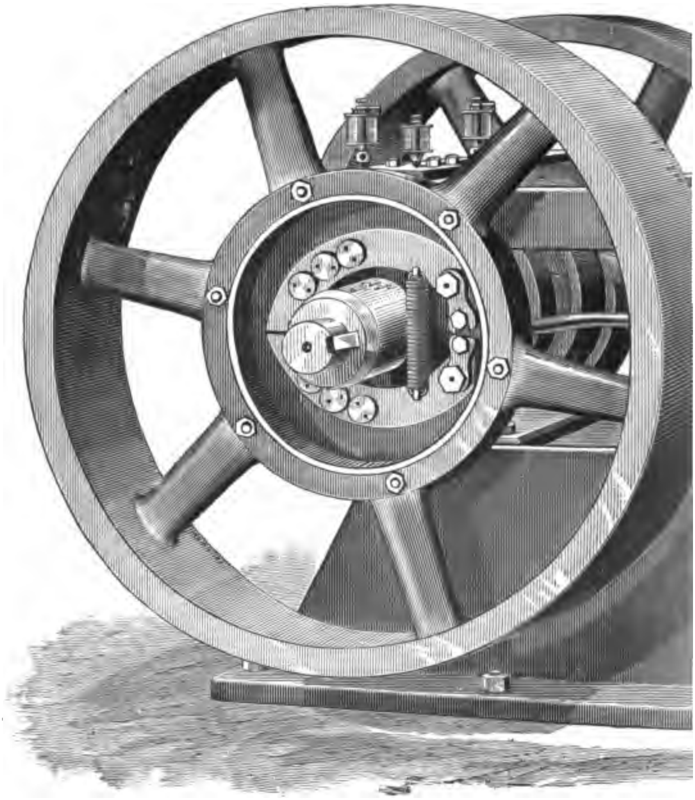
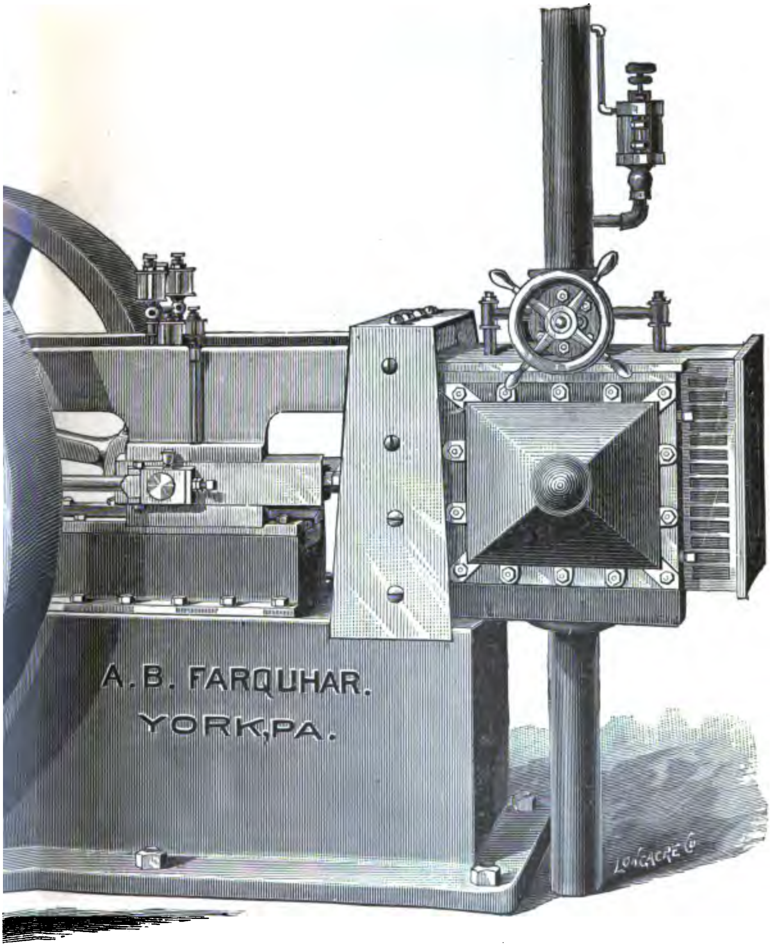


Fig. 5

A. B. FARQUHAR'S (YORK, PA.), STRAIGHT LINI


A. B. FARQUHAR.
YORK, PA.
LONGACRE Co

g. 59. Page 296

LINE, CENTRE CRANK AUTOMATIC HIGH SPEED.

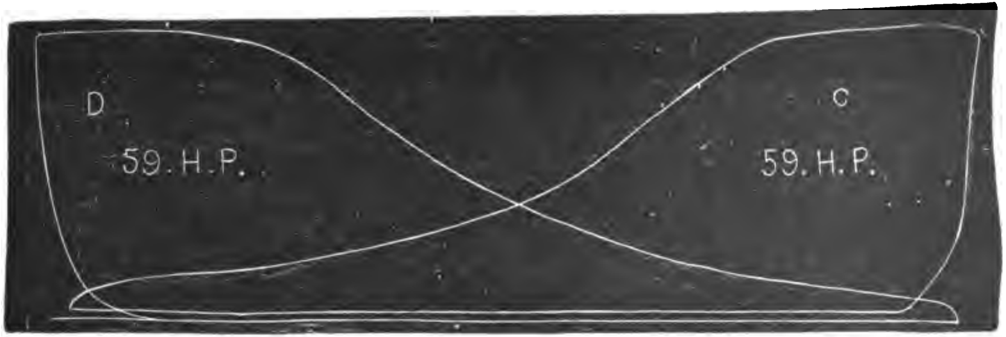


Fig. 60. Page. 296.

INDICATOR DIAGRAM FROM ONE OF FARQUHAR'S AUTOMATIC ENGINES.
(See engine, Fig. 59.)

speed at which they are run and the steam pressure. The steam pipes, valves and connections on all these engines are made large enough to supply steam for the engine at its highest rated power, and the engines are built with the expectation that they will be run at the maximum speed; all bearings being ample for high speed and long runs. Where circumstances make it desirable to use a slower motion, no harm is done by having an amply strong engine.

PRINCIPAL DIMENSIONS OF FARQUHAR'S AUTOMATIC ENGINES.

Horse power.	Cylinder in inches.	Fly Wheel, diameter in inches.	Face of fly wheel in inches.	Revolutions per minute.
20 to 40	8x 8	30 to 48	9	200 to 350
30 to 50	10x10	32 to 54	10	200 to 325
50 to 70	12x12	42 to 60	11	200 to 300
60 to 90	14x14	60 to 66	12	200 to 275
80 to 100	16x16	60 to 72	13	200 to 250
90 to 150	16x24	72 to 84	16	100 to 200
100 to 175	18x24	72 to 96	20	100 to 200
125 to 225	22x30	96 to 120	24	80 to 150

GENERAL DIMENSIONS OF HORIZONTAL RETURN TUBULAR BOILERS.

Heating surface estimated at about 15 square feet per horse power, our rating.

Horse Power	10	12	15	20	25	30	35	40	50	60	70	80	90	100
Diameter of boiler in inches	33	33	35	38	38	42	42	48	50	54	60	60	64	64
Length of boiler with extension smoke-box	8′ 4″	9′	9′ 5″	10′ 9″	12′ 6″	13′	14′ 3″	15′ 2″	15′ 8″	17′ 4″	17′ 4″	17′ 3″	17	17.6
Length of boiler for full front	7′	8′	8′	9′ 4″	11′	11′ 3″	12′ 6″	13′ 4″	14′	15	15	15	15	15′ 6″
Number of tubes.	24	24	30	32	36	42	43	48	52	60	68	59 4in. or 90 3in.	65 4in. or 98 3in.	65 4in. or 98 3in.
Diameter of tubes in inches	3	3	3	3	3	3	3	3	3	3	3	3 or 4	3 or 4	3 or 4
Length of tubes	7′	8′	8′	9′ 4″	11′	11′ 3″	12′ 6″	13′ 4″	14′	15	15	15	15	15′ 6″
Height of dome in inches. .	20	20	20	24	24	24	24	24	30	30	32	32	32	32
Diameter of dome in inches	20	20	20	24	24	24	24	24	30	30	36	36	36	36
Length of smoke-stack in feet.	24	24	28	28	35	35	40	40	40	40	50	50	60	60
Diameter of smoke-stack in inches.	9¾	12	15	18	19	21½	21½	24	26	28	30	32	34	36
Weight of plain boiler . . .	2700	3100	3630	4000	4550	5000	5500	6300	7600	8900	9400	11600	12500	14800
Weight of boiler with all castings	3400	3800	4430	5100	5600	6500	7100	8000	9400	10900	11000	14800	15500	17800

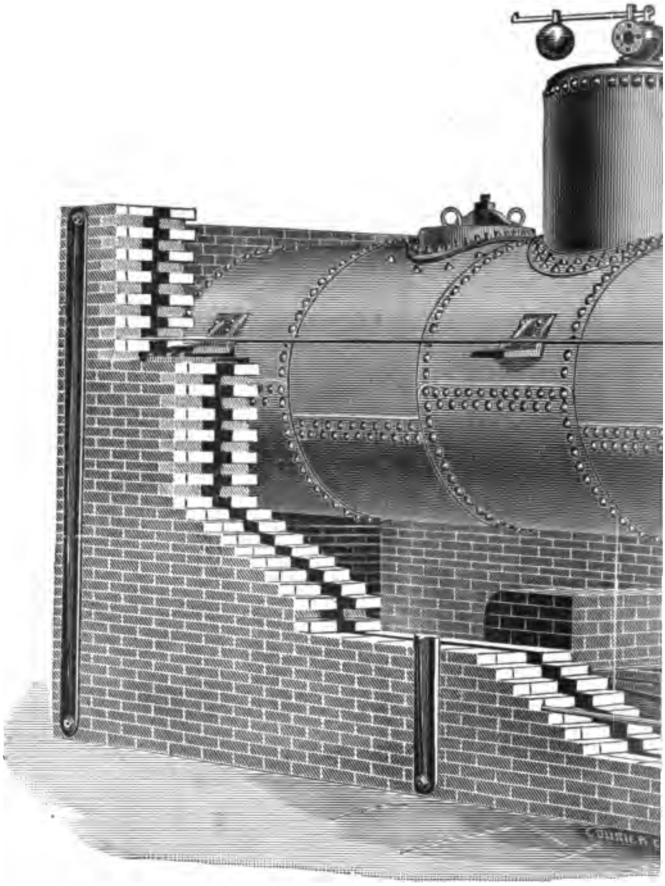
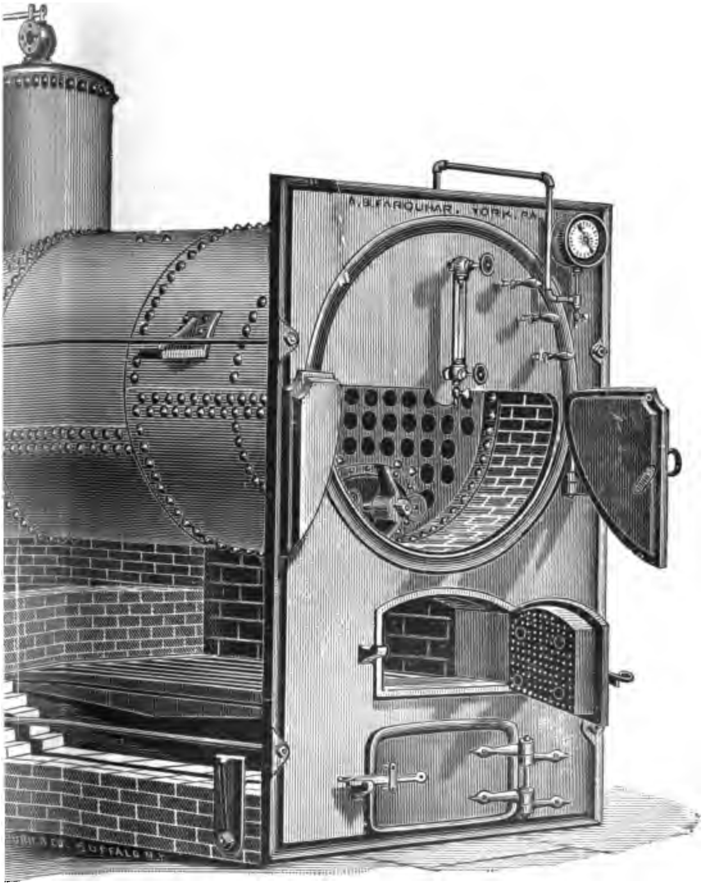


Fig. 61.

HORIZONTAL TUBULAR BOI


A. B. FARQUHAR, YORK, PA.
& CO. BUFFALO N.Y.

61. Page 298.

AR BOILER WITH FULL FRONT.

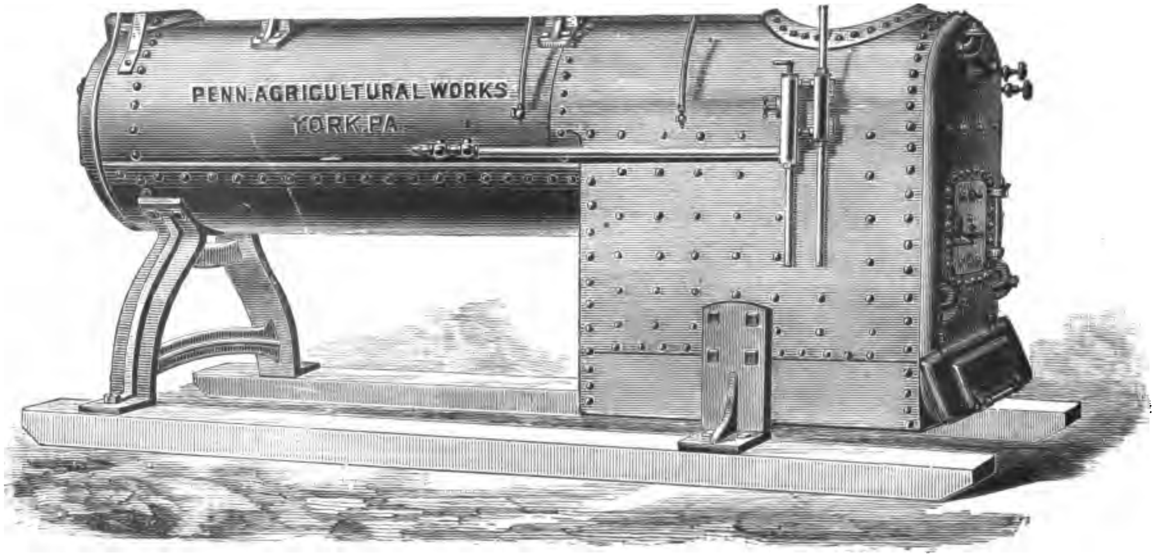


Fig. 62. Page 299.

LOCOMOTIVE OR FIRE-BOX BOILERS.

Portable, with Charcoal-iron or Steel Shells, and Steel Fire boxes.

LOCOMOTIVE OR FIRE-BOX BOILERS PORTABLE WITH CHARCOAL-IRON OR STEEL SHELLS, AND STEEL FIRE BOXES.

Fig. 62 represents the most approved style of locomotive boiler. It is the custom with some manufacturers to extend the fire-box under the grate, as the boiler is easier to make in this way, requiring no frame at bottom, but experience proves this addition of no practical service and very objectionable, as it becomes a receptacle for sediment.

Horse Power.	3 to 4	4 to 5	5 to 6	6 to 8	8 to 10	10 to 12	12 to 15	15 to 18	18 to 20	20 to 25	25 to 35	30 to 40	35 to 45	40 to 50	50 to 60
Waist Diameter in inches	24	24	26	26	26	29	33	36	36	39	39	42	42	48	48
Length of Fire Box in inches.	29	29	29	31	37	40	41	49	49	51	51	54	54	54	60
Width of Fire Box in inches	20	22	22	22	22	24	28	31	31	33	34	37	37	43	44
Height of Fire Box in inches.	25	27	27	28	28	29	30	32	32	39	40	46	48	48	54
Number of Tubes. .	22	22	24	26	32	36	26	28	30	32	36	46	46	53	60
Diameter of Tubes in inches	2	2	2	2	2	2	3	3	3	3	3	3	3	3	3
Length of Tubes in inches.	44½	50	52	55	63	68	76	93	96	103	117	108	115	112	132
Diameter of Smoke-stack in inches . .	8	8	8	9¾	9¾	9¾	12	15	15	18	19	19	21½	24	26
Length of Smoke-stack in feet . . .	6	7	7	8	10	10	12	16	16	18	18	20	22	30	36
Estimated Weight. .	1000	1150	1400	1600	1900	2400	3300	4200	4500	4800	5600	6000	6500	7300	8500

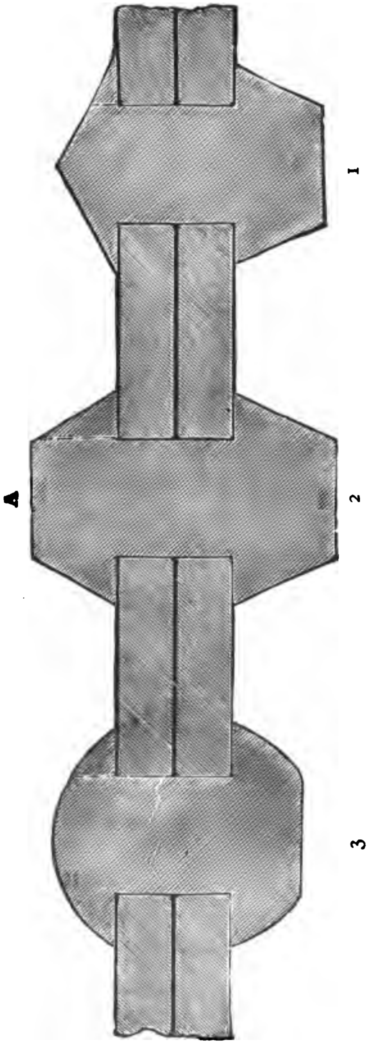


Fig. 63.

At a meeting of the Master Mechanics' Association of 1881, Button-set Riveting was unanimously approved. The sample of sheets riveted together by different methods and cut to exhibit vertical sections showed that the button-set method was superior to all others known to them at that time. Clinch's Patent Rivet-set (see example No. 2 riveting above) has been approved by all master mechanics who have examined it. The rivet-head A is an exact duplicate of the original head, which is the strongest known. A comparison of the vertical line in head A with other heads will demonstrate its greater strength. No. 1 is a sample of hand-riveting, and No. 3 of the old style button-set riveting

CHAPTER XVII.

IMPROVED CORLISS ENGINES. BY THE WATTS-CAMPBELL COMPANY, NEWARK, N. J.

Strength and Design.

EACH size of these engines is made from its own special patterns, which makes it possible to maintain *correct proportions.* In the design of the several sizes, if they have erred at all it has been in the direction of unnecessary strength rather than in the opposite direction. This, at the most, can only result in a little additional cost of material and labor, and can be counted *but once;* but an error in the opposite direction is a *constant cost* to the purchaser, in decreased efficiency, expensive break-downs and delays, and inferior durability.

For a given cylinder capacity these engines are heavy, but we are satisfied that any reduction in this respect would be a direct loss to the purchaser, even if the price were regulated by the weight. They sell an engine to work at a given power; they are aware that circumstances are likely at any time to require this rated power, either temporarily or permanently, to be materially exceeded —.perhaps doubled. An engine should always be so constructed that this may be done without danger of breakage or heating. An engine is not cheap at any price unless it possesses sufficient strength and ample rigidity to permit this.

Materials and Construction.

In the construction of these engines they use none but the material best suited to each particular detail. The *cylinders* are made from special brands of iron, mixed and remelted in their own foundry, for the purpose of making a perfectly homogeneous casting of the desired quality as to hardness, and without blow-holes or other disfiguring and damaging features. They invite special attention to all the *castings* used in their engine-work, both as to smoothness and soundness as well as to strength.

For *forgings* they use either steel or iron, as experience has shown us to be best for the particular part for which the piece is used.

Their business is to build steam-engines, for doing which they are supplied with tools especially adapted to that purpose. The use of *jigs* and *gauges* enables them to preserve sizes and to manufacture the various parts in duplicate.

They will not, under any circumstances, vary from their fixed plan of using only the best material, and maintaining standard excellence of workmanship. *They will not build so-called cheap engines.* In this respect they study the best interests of their customers, as well as their own. Every engine sold by them is made for hard work, twenty-four hours a day if required.

RELEASING-GEAR.

This is shown somewhat in detail in Fig. 64. In the gear ordinarily employed with the Corliss engine, the construction is such as to lead to some inconvenience and annoyance from the parts wearing loose and shaky. The effect of the action of the releasing-gear is also to slightly disturb the governor; the latter is the more serious fault, because its tendency is to impair the regulation, preventing its being brought to that degree of perfection which is desirable.

The improved releasing-gear effectually overcomes these objections. The shock upon the governor incident to the use of the crab-claw is avoided, the effect at the time of release being unnoticeable. It is noiseless in operation. The arrangement of parts is such that the leverage to the detaching-gear is *absolutely uniform* under all conditions, and the work of unhooking reduced to a minimum; at the same time, a long and ample bearing-surface is provided to the moving parts, with the power applied substantially in line with the resistance. By the use of this releasing-gear they improve the regulation, avoid the necessity for any resisting attachment to the governor, and increase the durability very materially.

DASH-POT.

They have also recently added a new dash-pot. This is shown in Fig. 65. They consider

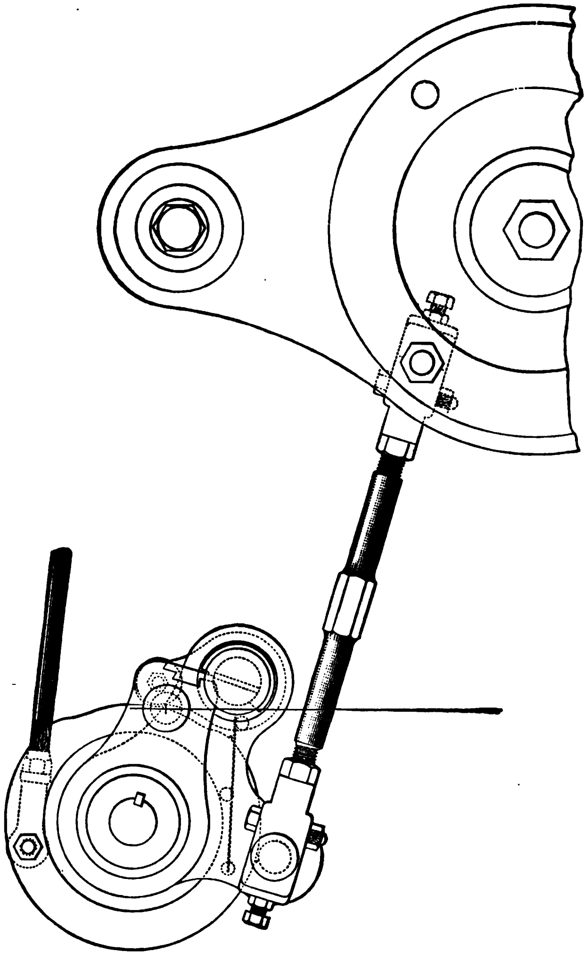


Fig. 64.

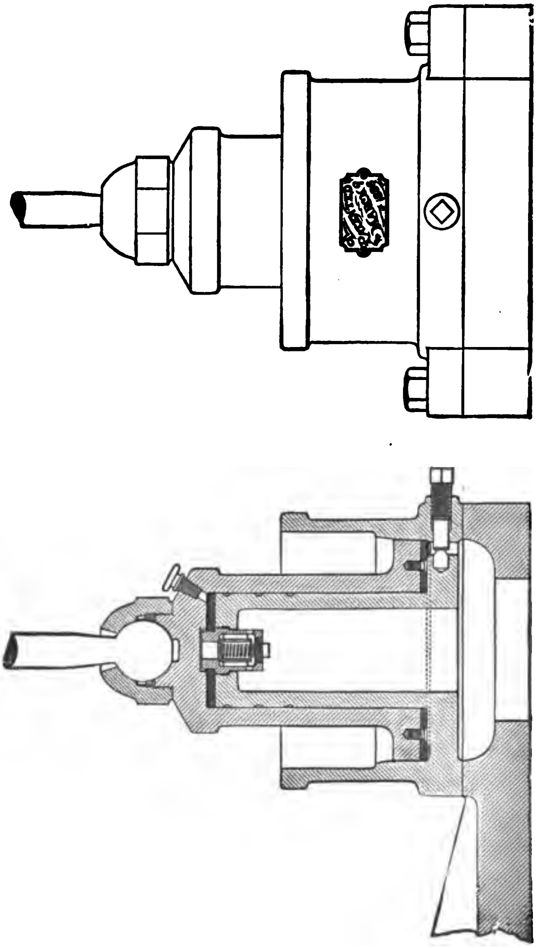


Fig. 65.

it in operation to be superior to any they have previously used for the purpose. By proper proportion they secure extremely rapid valve closure without jar, shock, or noise.

The vacuum which serves to close the valve is maintained in the chamber above the central post. As the piston descends, closing the steam valve, any small quantity of air that may have found its way into this chamber is displaced through the antomatic valve shown in the top post.

The cushioning is accomplished in the annular chamber at the bottom. The piston in falling is first partially obstructed in the tapered upper parts of the annular chamber; then, as it passes this tapered portion, it is more completely resisted, the only escape for the imprisoned air being such as is provided by the adjusting screw. By means of this screw, any desired adjustment of cushion can be made, interposed leathers preventing the parts from striking metal to metal while making such adjustment, or at any time while in operation.

BALL-AND-SOCKET-BEARING.

It will be noticed that the attachment of the dash-pot piston to the rod is by means of a ball-and-socket-bearing. This permits the piston to turn freely on the central post, thus promoting uniformity of wear, and increasing the durability of the parts.

In addition to permitting revolution of the piston, the ball-and-socket connection compensates for any slight fault in alignment, avoiding all danger of binding; it also forms an oil reservoir, from which the oil cannot leak, thereby insuring perfect lubrication.

The oil-hole in the top of the piston provides for conveniently oiling the other parts which move in contact.

ENGINE-FRAME.

The Girder Frame of this engine is of heavy weight, and carefully designed to resist the strains to which it is subjected. We believe that, with judicious distribution of metal, this type of frame is for a given weight the strongest, as it is certainly the most convenient, of any yet devised. Its peculiar elements of strength have sometimes led to such attempts at economizing in the use of metal as to reduce below proper limits the qualities of strength and stiffness.

An examination of any of the sizes of these engines will satisfy the most critical that no mistake has been made in that direction. The cost of the extra metal in an engine-frame, necessary to put its rigidity beyond question, is but a trifle compared with the advantages of that rigidity.

Unfortunately, an engine-frame may be of no more than half the weight required for stability, without much danger of breakage. But with the light frame the inevitable springing throws the parts out of alignment at every stroke of the piston, resulting in unnecessary friction and destructive wear, and frequently in breaking some of the working parts.

These frames will stand up squarely to work under any steam pressure employed in practice, and under loads twice as great as their rated power.

With the Girder Frame, if the work is properly done, the aligment is perfect, and with ample wearing surfaces it will remain so indefinitely.

EXHAUST PASSAGE AND CYLINDER.

These were the first builders of this class of engine to recognize the importance, in an economical point of view, of keeping the exhaust steam separated from the walls of the cylinder. Their views were that this would result in increased economy, in that the exhaust steam when passing in contact with the cylinder robs the latter of heat, which must be restored by the admission of a greater quantity of steam from the boiler to do a given amount of work. This view they have demonstrated to be correct, and in all sizes of their engines the exhaust from each end of the cylinder goes to a common outlet at the centre, through passages entirely distinct from the cylinder proper.

CONNECTING-ROD.

Their connecting-rods are made with gib-and-key adjustment, which they find to be the most reliable. The standard length of their rods is six cranks. This, though somewhat greater than usually found, they consider is enough better to warrant the increased cost.

The evil of short connecting-rods in creating unnecessary friction, and increased wear and strains, is not fully appreciated. The increased friction due to short rods is a constant cost in coal, and shortens the life of all parts of the engine.

Beyond a certain length of rod further increase will result in but little saving, will unwarrantably increase the cost of construction, and introduce inconvenient features; these facts clearly point to a limit. This limit these builders have fixed at six cranks, or three times the length of stroke.

CROSS-HEAD.

The cross-head is tapered and secured by a cross-key. This method they prefer to the screwed end, from the use of which serious accidents have occurred.

Although by the taper wedges and screws the most minute and accurate adjustment of gibs can be made, when properly adjusted the whole has the rigidity of a solid piece of metal. There are no parts liable to work loose and cause delay and accident.

The gibs are lined with best anti-friction metal, and present large wearing surfaces, backed up against the possibility of springing, which is sure to result with overhanging ends.

PISTON AND PISTON-PACKING.

The weight rests upon the centre ring, to which the piston and follower are securely attached. When by wear of the bottom of the centre ring, and of cylinder, the piston gets below the centre, it can be accurately centred by means of the adjusting screws. This they consider absolutely essential in a horizontal engine, in which, owing to gravity, the bottom of piston and cylinder will be subjected to somewhat the most wear. As this occurs, unless there are means for adjustment, the piston-rod will wear out of truth, the stuffing-box will be worn down, and a strain thrown on the rod liable to result in its fracture, or in breakage of some of the connecting parts.

The centre ring carries the weight of the piston, and protects the head and follower from wear.

It is necessary, for safety, that the centre ring be somewhat smaller than the bore of the cylinder. If turned round, as is ordinarily done, when placed in the cylinder, the bearing will be only in a line on the bottom. This will, from wear on a small surface, soon not only locate the piston below the centre, but will render cutting or scoring probable.

By the method of these builders of turning the centre ring, the lower or bearing part is made to exactly fit the bore of the cylinder, the ring being turned out of round to give the requisite clearance. This gives full bearing surface, from the start.

The packing consists of two small rings, one at either edge of the centre ring. These are turned somewhat larger than the bore of the cylinder, then cut, and halved together at the joints. When in place, they keep in easy contact with the cylinder, without undue friction, compensating for wear by their own elasticity.

Light springs are supplied, which assist in keeping the rings in contact with the cylinder until they are worn out. In practice they find that this packing requires no attention except, perhaps cleaning once in about six months, at which interval the piston should be centred in the cylinder.

This piston is simple in construction, the packing is self-setting, and the piston can be kept where it belongs, viz., in the centre of the cylinder.

SIMPLICITY.

The claim is sometimes made that the Corliss valve-gear is complicated in its detail, and not easily comprehended except by experts. The direct contrary of this is true of the valve-gear of these engines. The motion and the arrangement of parts are remarkably simple, and can be understood by any one of ordinary intelligence from the observations of a few minutes.

All the parts are in plain sight, and can be observed at any time while the engine is in motion. There are absolutely no adjusting parts inside, where they may get loose or out of position without being seen, or be subject to corrosive action, and hence sudden failure.

In all other respects, as well as in the valve-gear, their engines are simple, having no parts that cannot be replaced or repaired without special tools or appliances. Such special tools as they use in the manufacture are used for convenience and for expediting the work, and not from necessity. These engines can be repaired in all of their parts should it be necessary, by any machinist of fair ability, and in any machine-shop provided with the ordinary tools of the trade, while any engineer who is competent to take care of an ordinary slide-valve engine will have no difficulty in running one of these engines.

Approved means of adjustment for wear are provided wherever required, as well as ample means of lubrication.

ADAPTABILITY.

These engines are adapted for all ordinary purposes where steam-power is required. Their perfect regulation and strong and durable construction make them particularly desirable for electric-lighting purposes, for driving the machinery in cotton and woolen mills—in fact, for all uses where good regulation and uninterrupted service are essential; while their simplicity and economical use of steam recommend them to all steam-users who study economy in connection with their motive plant.

SELECTION.

On p. 325 we give table of sizes, which will be found of service in the selection of an engine of suitable dimensions for the work it has to do.

It is hardly necessary to offer any argument in favor of an engine that will work with good economy of steam consumption. It has come to be universally conceded that an economical engine is the cheapest for nearly every purpose; this without much reference to the cost of fuel. What is saved in the difference in price between a good and a "cheap" engine must be expended in paying for extra boiler capacity to furnish sufficient steam, and it would be expended the second time, in a few years, in keeping the extra boilers in repair, and in renewing them.

HORSE-POWER.

In making a selection, the horse-power required is the first thing to be decided. Having settled this, then from the table on p. 325 the size can be selected. It will almost always be advisable to take the horse-power from the first or lowest rating. The difference in economy will be scarcely appreciable if the work required falls anywhere between the lowest and the highest ratings. If the load is materially less or greater than these ratings, respectively, the economy will not be so good.

The tendency is almost invariably to add to the load of an engine by adding machinery to the plant; hence, if an engine is selected that will do the work at the lowest rating, it will do this work with first-rate economy, and the load may be increased to the extent indicated by the difference in the two ratings, the economy continuing good. We urge this view upon all purchasers of steam engines. It will be seen that when a selection is made upon the basis indicated, the economical range, as *by the table*, is quite large; and this can be still further increased as follows: It will be noticed that the rating is with quite low initial pressure—seventy pounds. Increasing this will increase the power.

Furthermore, this rating is from a very moderate number of revolutions, which can be increased with corresponding increase of power, and still be far within the limits of danger from heating and accidents from high speed.

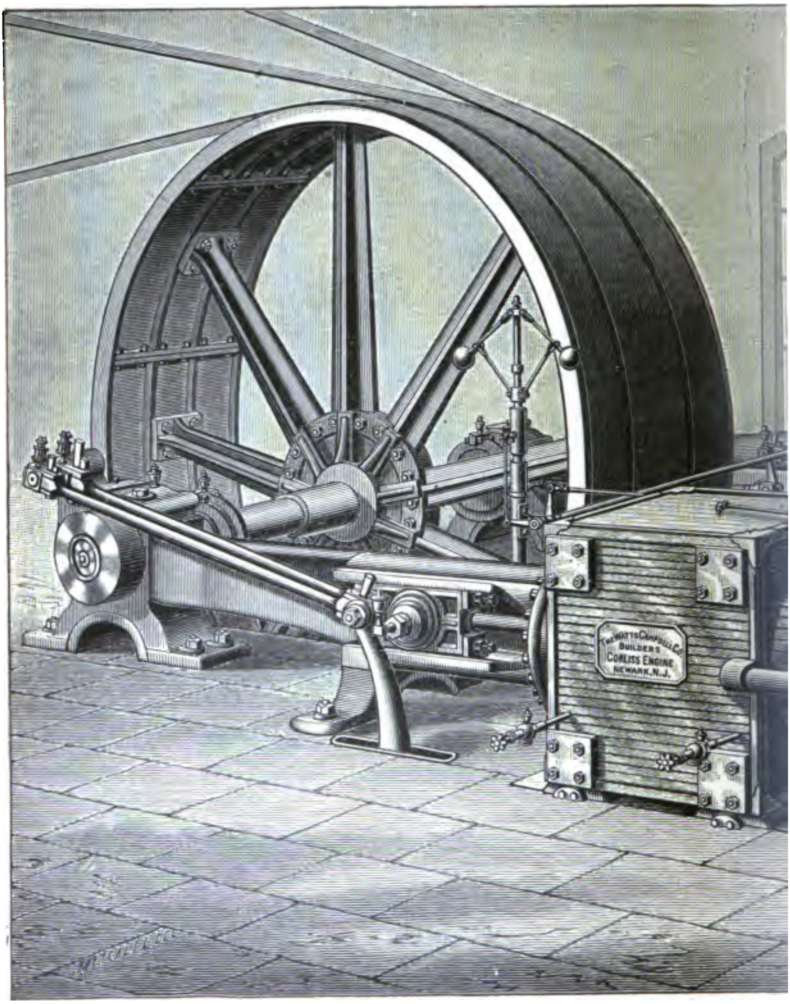
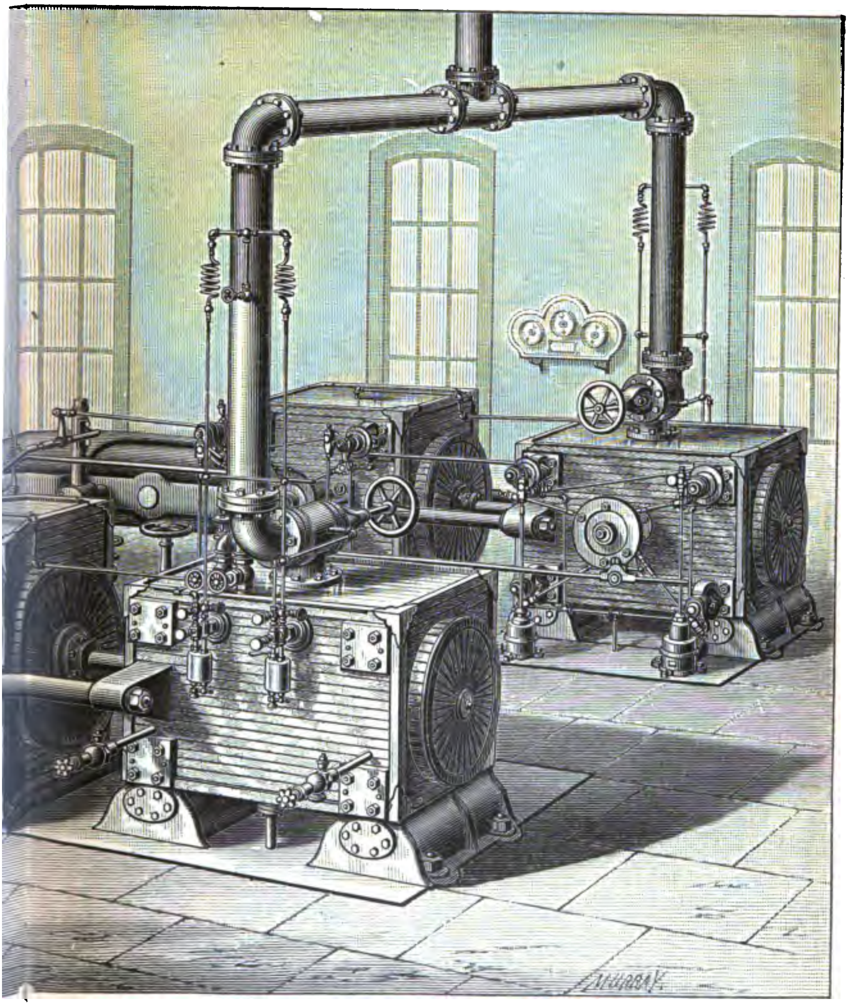

THE WATTS CAMPBELL CO.
BUILDERS
CORLISS ENGINE
NEWARK, N.J.

Fig. 6

COMPOUND ENGINES



66. Page 317.

S AT THE SHREWSBURY MILLS.

TANDEM COMPOUND CONDENSING ENGINES AT THE SHREWSBURY MILLS.

Fig. 66 represents a pair of compound engines recently put in the Shrewsbury Mills, at East Newark, N. J. These are new mills, devoted exclusively to the manufacture of linen thread, and excellently arranged in all their appointments. The engines were erected by the Watts-Campbell Co., of Newark, N. J., and the guarantee required makes their performance of unusual interest to engineers and steam-users.

The engines are tandem compound, coupled to the shaft at right angles; an excellent arrangement for securing regular motion and smooth running. The high pressure cylinders are 20 inches diameter and the low pressure 36 inches; stroke of pistons, 48 inches. The engines run at a speed of 64 revolutions per minute. Both the high and low pressure cylinders are steam jacketed, the former with steam direct from the boiler, and the latter with the exhaust steam from the high pressure cylinders; this makes the jacket temperature in each case equal to the initial temperature in the cylinders, respectively. The cylinders and heads are also very thoroughly protected from loss by radiation by being first covered with asbestos, then hair-cloth, and lastly lagged with wood. The construction is also such as to leave but very little surface that cannot be so protected.

The exhaust from the high pressure cylinder passes down through the legs to the receiver, which is cast as part of the low pressure cylinder and includes the jacket space of that cylinder. From the low pressure cylinder the exhaust goes through a large rectangular passage to the condenser, which is situated midway between the two low pressure cylinders. A small pump returns the water of condensation from the jackets to the boilers.

But one air pump is employed, which is driven by a return rod from one of the crank-pins. There is nothing peculiar in the construction of the air pump, it being of the single-acting type used by the Watts-Campbell Co. with all their condensing engines. The arrangement of the passages, however, both from the low pressure cylinders to the condenser and from the condenser to the pump, are well designed to avoid any obstruction to the passage of steam and water.

The main shaft is 16 inches diameter at the center or wheel fit, and 13 inches at the journals. The band fly-wheel is 25 feet in diameter, built up in ten segments. It has a face of 6 feet 2 inches, turned for two 28-inch belts and one 10-inch belt. The weight of the fly-wheel is 73,000 pounds. This is a heavy wheel for the power of the engines, and especially for a pair of engines coupled at right angles. This heavy fly-wheel has doubtless something to do not only with the almost perfect uniformity of motion, but as well with the smooth and noiseless running.

The valve gear is of the Corliss type, with the modifications that have been introduced from time to time by the Watts-Campbell Company —some of which modifications are important in their effect upon the operation of the engine. The speed is controlled by means of a small fly-ball governor, running at very moderate speed; the governor looks entirely inadequate to control admission by eight steam valves, but it does this with great precision, and without the use of a dash pot or equivalent attachment to prevent fluctuation; in fact, the absence of intention of the governor to fluctuate is remarkable, considering its small size and the number of valves it controls, and shows that it has literally no work to do. This absence of shock to the governor is mainly due to the excellent action of the releasing gear, the invention of John Young, who has for years been connected with the firm. The dash pots used are also the invention of Mr. Young, and the perfect action of these, with the easy action of catching and releasing the valves, makes the action so nearly noiseless as to leave nothing further to be desired in this respect. The governor is connected to a cross-shaft from which small single rods extend to the releasing mechanism of the four cylinders, doing away with the use of the double rods usually employed. An illustration and description of this releasing gear and of the dash pot, both of which have proved in use even more effective and satisfactory than it was believed they would, appeared in the *American Machinist* of June 26, 1886, together with other details.

The Watts-Campbell Company, in their compound engines, adhere to their established practice in simple engines of making the connecting rod six cranks in length—a practice which we believe they are wise in maintaining. The piston rods have two different diameters in their length, the difference being sufficient to afford a taper seat for the low pressure pistons. These pistons are held in place by a key. By disconnecting the rod at the cross-head and moving the low pressure piston back into the space between the two cylinders the key can be removed, then by moving the rod forward the piston can be removed.

A commendable feature in the construction of these engines is in the fastenings which hold the bed plates to the pillow block. In addition to the usual bolts, which are from necessity some distance from the line of strain, a recess is cast in the front side of pillow block and a corresponding recess is cast in the front side of the frame against the pillow block. Then a wrought iron link is shrunk over the parts enclosed by the recesses, binding the pillow block and frame firmly together at a point the nearest practicable to the line of strain.

The boilers that supply the steam were not furnished by the builders of the engines. They are of the common return tubular type, 6 feet in diameter and 16 feet in length, set in the ordinary manner. There are six of these boilers, steam being required for various purposes in the mill. At the front and above the boilers is a steam drum 24″ diameter, extending entirely across the boilers and to which they are connected by pipes and stop-cocks. From this drum a 12″ steam pipe extends to the branch pipes for the engines, the branch pipes being each 7 inches in diameter. These large steam pipes and the long steam ports to the cylinders, the latter extending as far as the jackets, or very materially beyond the bore of the cylinders, with the arrangement of valve gear which gives a full port when the piston has moved one-eighth of its stroke, provide an ample passage for steam to the cylinder, a fact that will be clearly seen by an inspection of Fig. 66.

Reference has been made to the terms of the contract. According to this the engines were guaranteed to develop 700 indicated horse power with boiler pressure of 110 pounds and on a coal consumption of $1\frac{7}{10}$ pounds per indicated horse-power per hour. Upon starting the engines it was found that it would not, at least for some time, be practicable to load them to more than about 300 horse-power; it was then concluded to disconnect one of the pair and test the other, the builders of the engines waiving the right to steam of 110 pounds pressure and using but 80 pounds; two boilers only were used. While the engine was run only through the ordinary working hours—ten and one-half—all the coal used during the twenty-four hours was charged against it; this included coal for banking fires, getting up steam in the morning, etc., in other words, all the coal weighed into the fire room, without rebate of any kind. The test was under the supervision of Thomas Gardner, chief engineer of the Clark Thread Mills at East Newark, who acted for the Shrewsbury Mills, and John Young, who acted for the Watts-Campbell Company. The firing was done by the regular fireman. In every way it appears to have been a thoroughly practical and reliable test, under the conditions that would exist in the every-day operation of the engines. The test was continued for four days—96 hours—a large number of diagrams being taken from which to compute the power. The table shows the result.

RESULT OF A FOUR-DAYS' TRIAL OF THE WATTS-CAMPBELL CO.'S COMPOUND TANDEM ENGINES AT THE SHREWSBURY MILLS,

Commencing May 3d, 7 a. m., and ending May 7th, 7 a. m.

	Lbs. coal used in 24 hours.	Running time.	Coal used per hour running time.	Ind. H. P.	Coal per hour per 1 H. P.	Rev. per min.
May 3d. . .	5000	10.5 hrs.	476.19	273.09	1.74	64
May 4th . .	5500	10.5 "	523.81	295.48	1.77	64
May 5th . .	5700	10.5 "	542.86	311.74	1.74	64
May 6th . .	5393	10.5 "	513.62	309.81	1.65	64
Average	. .	. . . .	514.12	297.53	1.73	.

Constant every-day run. No coal deducted for banking fires. No allowance for ashes.

This is remarkably low coal consumption under the conditions of the test. From this, to make the test conform to ordinary practice, the coal consumed in banking fires and getting up steam should be deducted. This was not done, because there was no necessity for going into such a calculation; it was evident that the engine had materially exceeded the duty required by the contract. The allowance to be made could not, we should think, be less than 400 pounds for each boiler, or 800 pounds altogether. Assuming this to be correct, the coal (with no deductions for ashes and dirt) would be 1.46 pounds per indicated horse-power per hour. What the result would have been with steam of 110 pounds pressure can only be conjectured.

The gauge pressure, corrected, was 80+ pounds, vacuum 27 inches. There was but very slight difference between the back pressure in the high pressure cylinder and the forward pressure in the low pressure cylinder previous to cut-off, as well as between the back pressure line in the low pressure cylinder and the pressure in the condenser, as represented by the vacuum noted. That all these differences are so slight is due, of course, to providing very large steam and exhaust passages, and to controlling them properly.

Altogether the performance of the Shrewsbury Mills engines is a strong argument in favor of compounding. And the excellent steam plant of these mills may be taken as a model in its way.

The growth of the compound engine in this country has been slow, but it is now gaining ground very rapidly, a fact that is mainly due to actual experience in its use. It has never been helped along to any extent by scientific considerations, but its progress will be all the more rapid because the arguments in its favor rest upon actual facts gained by experience.

HORSE-POWER AND PRINCIPAL DIMENSIONS.

Cylinder.		Band Wheel.				Horse-power. 70 pounds (by gauge) initial pressure. Cut off in parts of stroke.		Length from centre of crank shaft to back cylinder head.	Crank Shaft. Average Length.	From centre of crank shaft to foundation plate.
Diameter. Inches.	Stroke. Inches.	Revolutions.	Diameter Feet.	Face. Inches.	Weight. Pounds.	$\frac{2}{10}$	$\frac{3}{10}$	Feet. Inches.	Feet. Inches.	
12	28	80	9 to 10	13	6,500	42	52	12′ 8″	6′ 0″	1′ 9″
14	32	75	10 to 12	17	8,000	58	72	14′ 4″	6′ 6″	1′ 9½″
16	30	75	10 to 12	19	9,500	75	83	13′ 10⅛″	7′ 6″	2′ 1⅝″
16	38	70	12 to 13	19	10,000	80	100	16′ 10½″	7′ 6″	2′ 1⅜″
16	42	70	12 to 13	21	10,500	85	105	18′ 4⅛″	7′ 6″	2′ 1⅜″
18	36	75	12 to 14	23	12,000	100	125	16′ 2¼″	8′ 6″	2′ 3″
18	42	70	14 to 16	25	16,000	125	155	18′ 4⅜″	8′ 6″	2′ 3″
20	42	70	14 to 16	28	18,000	150	187	18′ 10½″	9′ 0″	2′ 6″
20	48	65	16 to 18	32	20,500	160	200	21′ 1″	9′ 0″	2′ 6″
22	42	70	14 to 16	32	23,000	175	225	19′ 3″	10′ 0″	2′ 6″
22	48	65	16 to 18	32	23,000	185	230	21′ 6½″	10′ 0″	2′ 6″
22	52	60	18 to 20	32	27,000	200	250	23′ 0¾″	10′ 0″	2′ 6″
24	42	70	14 to 16	38	27,000	215	270	19′ 6″	11′ 0″	2′ 11″
24	48	65	16 to 18	38	31,000	230	285	23′ 8½″	11′ 0″	2′ 11″
24	56	60	18 to 20	38	33,000	260	325	24′ 9½″	11′ 0″	2′ 11″
26	48	65		Wheels	as desired	275	344	22′ 6½″	11′ 0″	2′ 9″
26	60	60				315	390	26′ 11½″	11′ 0″	2′ 9″
28	54	60				330	410	24′ 8¾″	11′ 4″	2′ 10″
28	60	60				360	450	26′ 9″	11′ 4″	2′ 10″
30	54	60				380	475	25′ 2″	11′ 4″	2′ 10″
30	60	60				400	500	27′ 6″	11′ 4″	2′ 10″
32	48	65				420	530	23′ 1½″	13′ 6″	2′ 10″
34	60	60				470	600	27′ 7¼″	13′ 6″	2′ 10″
34	60	60				540	700	27′ 9″	14′ 9″	2′ 11″

CHAPTER XVIII.

STEAM YACHTS AND LAUNCHES.

HITHERTO the design and construction of a steam yacht has been carried on in a hap-hazard sort of a way, that is only too apparent from a casual inspection of the vessel when completed. One firm is called upon to whittle out a model of a certain length, another is asked to build the engine, while a third supplies a boiler of certain dimensions. The interior fittings are supplied by individuals, perhaps totally unacquainted with marine work. Each individual concerned may be competent to carry out his own particular part of the work, but he pays but little attention to any other portion, and the result, while perhaps correct in detail, is decidedly lacking in harmony.

The first requisite is speed, and accommodation comes next in the scale. The former can only be obtained by an easy model, and great power, while the latter demands close attention to the utilization of space. In order to produce an average of these essentials, the engine and boiler must be as small as possible, while at the same time capable of exerting great power, and so designed as to be free from vexatious breakdowns under continuous work. The consumption of fuel should be reduced to a minimum in order to economize space, and to avoid the inconvenience of frequent coaling.

In addition to these requisites a certain amount of *beauty* is necessary in order to harmonize with the purpose for which the vessel is to be used. The most important point in naval design is *harmony of details*, and this can only be attained by the influence of *one competent mind.* Errors in steam launches resulting from incorrect design are of comparatively less importance, but in steam yachts, where the amount of money involved is much greater, it is important that the design should be correct.

Fig. 67 shows the steam yacht "Juno," owned by J. R. Purchase, Esq., Minneapolis, Minn., and in use on Lake Minnetonka, Minn. Designed and constructed by Chas. P. Willard & Co., Chicago, Ill.

It seems needless to enumerate the purposes for which small steamers of this kind can be used; but we might mention their use on lakes and rivers for pleasure purposes and for the transportation of mails in districts where roads are very bad, or actually impassible, at many seasons of the year; also for ferry boats or for carrying fishing and hunting parties on lakes and rivers to fishing or hunting grounds, and particularly for towing barges laden with lumber, salt, coal, stone, or other products of the country, on lakes, rivers or canals. The uses to which such a boat can be put are greater than we can name here, and when once generally known they will find an introduction to all the small lakes

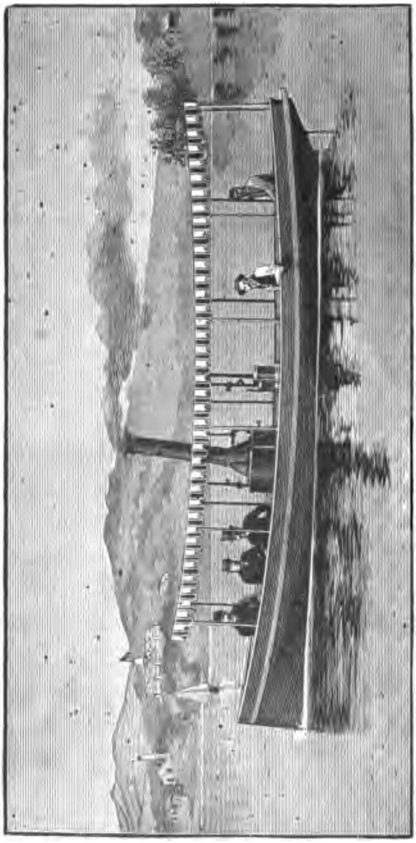


Fig. 67.
STEAM YACHT "JUNO."

and navigable streams in the country. As at present built they are so simple and easily managed that any person of ordinary intelligence can use one of them, and one man alone can fire and run a 30 or 33 foot boat without any assistance, provided the steering-wheel is placed just behind the engine.

Frequently a number of gentlemen associate together and buy a launch jointly for the use of their families, or several persons owning barges will club together and buy a small boat for towing all the barges. It is not uncommonly the case that one of these boats will earn its entire cost in one season, when used for hire on a small lake or river, where there are summer visitors.

YACHT BOILERS.

In selecting the proper form of boiler for a steam yacht due regard must be had to the form of the hull, and also to the kind of fuel to be used. It would not be advisable to use a vertical boiler of great power in a long, narrow hull, calculated for high speed, nor would it be well to adopt this form in localities where wood is the only available fuel.

Usually, however, the vertical boiler will be found most convenient for launch purposes.

SUBMERGED TUBE VERTICAL MARINE BOILERS.*

Where boats are required for use in waters that are under United States marine supervision, it will be necessary to have boilers that conform in every respect to the United States marine laws. These require, among other things, that vertical boilers used for marine purposes should have submerged tubes. The fact that a boiler has passed Government inspection is the strongest guarantee of the quality of the material and workmanship, as the requirements are extremely rigid.

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* Fig. 68.

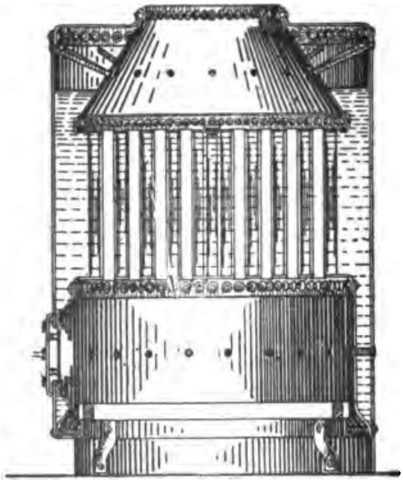

Fig. 68.

SUBMERGED TUBE VERTICAL MARINE BOILERS.

GENERAL DIMENSIONS OF YACHT AND LAUNCH VERTICAL MARINE BOILERS,

Designed and Constructed by Chas. P. Willard & Co., Chicago.

SHELL.		TUBES.			FIRE BOX.							
Diameter.	Height over Base Plate.	Number.	Diameter.	Length.	Height.	Diameter.	Height of Ash Pit.	Depth Cone.	Top Cone.	Square feet Heating Surface.	Size Engine Cylinder Suitable for	Weight in lbs.
22	36	33	1½	12	15	19	3	6	8	20	3 x4	330
26	48	43	2	18	18	22	4	8	14	43	4½ x5	580
30	48	68	2	16	18	26	4	10	16	60	5 x7	1040
36	50	87	2	18	18	32	5	12	16	92	6 x8	1190
40	53	99	2	18	18	36	6	14	16	100	6½ x8	1530
42	58	116	2	20	18	38	6	14	20	120	7 x8	1700
48	72	160	2	24	24	42	10	14	22	180	8 x 10	2600

The boilers with which launches are supplied should be so proportioned that they will stand very low in the boat, bringing the centre of gravity as low as possible. They ought to be found to be very quick and free-steaming boilers, and economical of fuel. These boilers should be built of charcoal hammered wrought iron or steel, and tested, by hydrostatic pressure, to 150 lbs. to the square inch.

Where required for navigation of waters under Government supervision, it will be necessary to have submerged tube vertical boilers, or the horizontal boilers.

No one thing connected with a steam launch is of more importance than to have the boiler free from all defects. Only the best quality of iron or steel should be used in the construction of it, as it is desirable, particularly with small boats, that the working steam pressure should be as high as possible.

The two forms of launch boilers which we illustrate in this book will all be found to steam freely, without the very objectionable feature of priming. This difficulty is frequently encountered in small boilers used in launches, particularly in rough water, where the disposition to prime, in small boilers of faulty construction, when crowded up to their full steaming capacity, is aggravated by the slopping over of the water from the rolling of the boat, making it difficult to get dry steam.

Probably no form of horizontal boiler used in launches combines in so great a degree, strength, compactness and low centre of gravity, as the form of boiler illustrated herewith, which is called the "Clyde" or Scotch boiler. Fig. 69.

As will be seen by the figure, all the surfaces of the fire-box and the shell of the boiler are cylindrical, this being the strongest form in which the material can be put together. The fire-grate is placed in the large central tube, and the fire is entirely surrounded by water. The "Clyde" is a return-tubular boiler. The fire passes through the large central tube into the

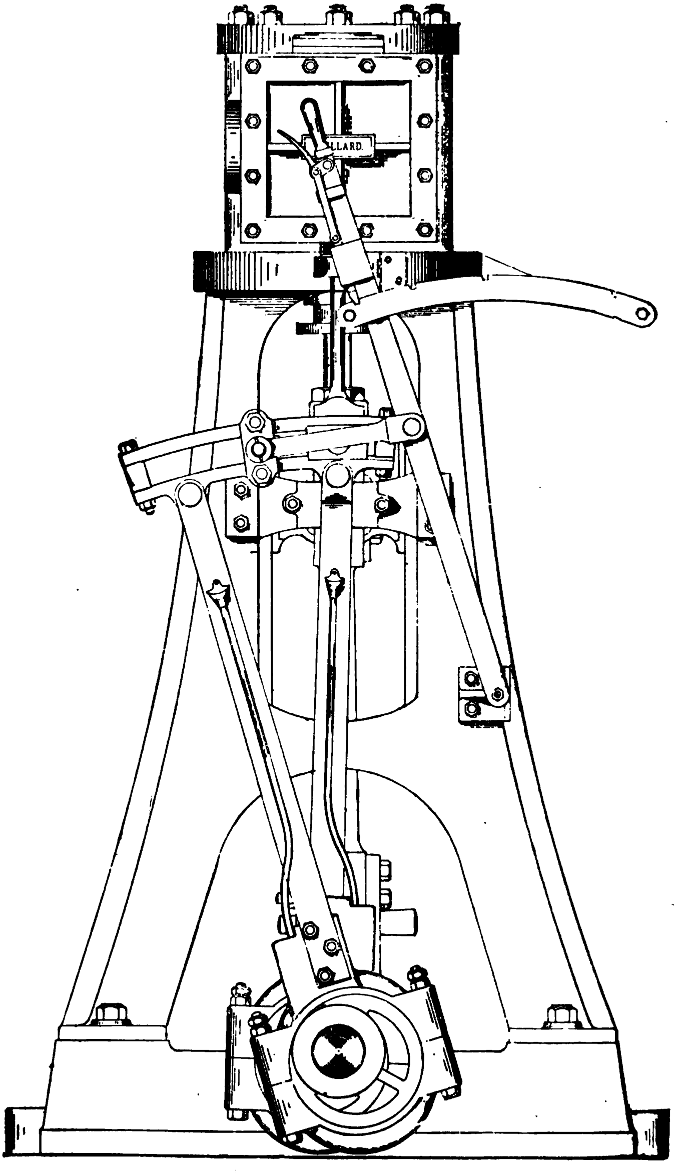

Fig. 70. Page 333.

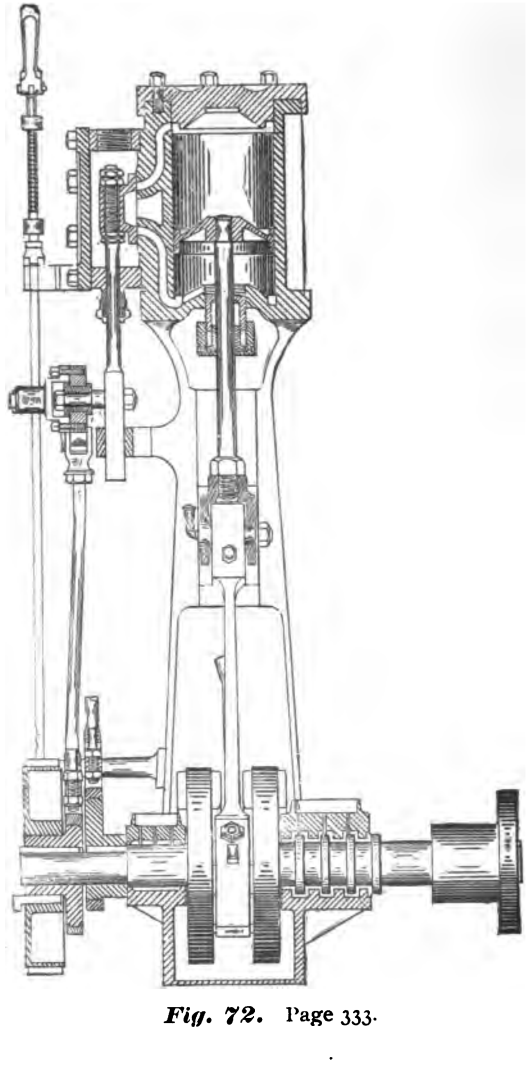

Fig. 72. Page 333.

combustion chamber at the rear of the boiler, and returns through the smaller tubes to the smoke bonnet, at the front end of the boiler, thence into the funnel, thus giving the greatest possible economy in the use of fuel. The tubes are, at all times, below the water line.

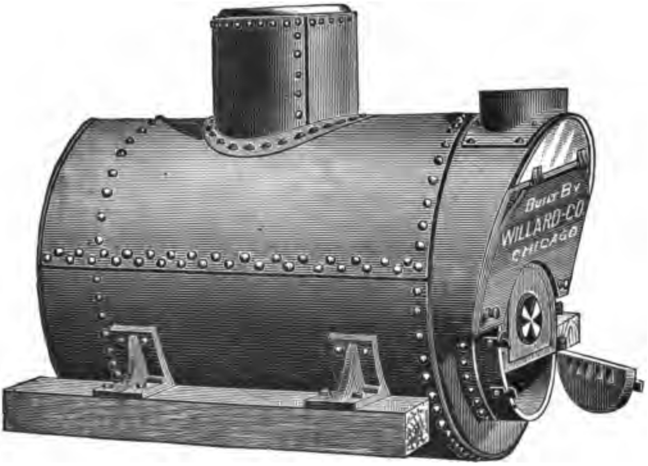

Fig. 69.

THE CLYDE BOILER.

THE ENGINE. Figs. 70, 71, 72.

The cylinder and frame, and what is usually called the bed-plate, form one casting, the frame forming a hollow column with suitable openings for access to the moving parts.

The piston is conical in form, thus insuring a maximum of strength with a minimum of material. The cross-head is of bronze, with the piston rod threaded into it and secured by a jam-nut. The connecting-rod is rectangular in cross section, having a solid end at the cross-head, and brass boxes fitted with a wedge for taking up the lost motion. The crank-pin end of the rod is provided with the usual strap and key. All the journals are of ample size, thus insuring cool running even under severe conditions.

The link motion is of the Stephenson type, and is designed so as to produce an equal cut-off for all points of the stroke. All the small parts, as far as practicable, are of steel, as light as is consistent with required strength.

The crank-shaft is of steel, the crank-arms being fitted with counterbalances eccentrically bored to fit crank-discs, and firmly held by bolts. The after journal forms the driving bearing, the thrust of the screw propeller being taken by four steel rings, solid with and projecting from the shaft, providing ample surface, and insuring freedom from lateral action.

In ordinary operation, these engines are run at a speed of 300 revolutions per minute, with the steam cut-off at half stroke. At this speed there is little vibration, the weight of the reciprocating parts being accurately counterbalanced.

Any lateral movement of the cylinder is largely avoided by the suspended position of the shaft. The centrifugal action of the counter-balance, which is a trifle greater at midstroke than that of the connecting-rod, has a tendency to impart a lateral movement to to the shaft. This action has been carefully considered, and the foundation timbers so located as to resist this tendency.

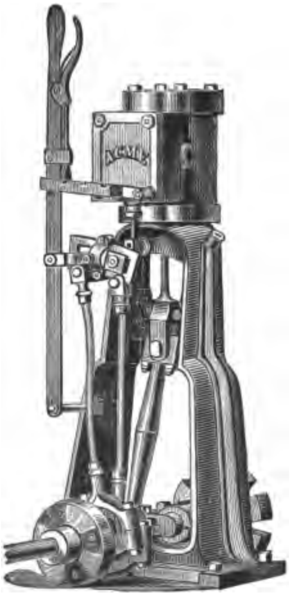

Fig. 73.

THE "ACME" LAUNCH ENGINE.

The "Acme" launch engine shows the smallest size marine engine. It is 3 inch bore, 4 inch stroke, and will develop, if required, about four indicated horse-power. The crank-shaft, piston-rod, valve-stem and pins are made of steel as well as the link. It is a very quick, powerful little engine. Its weight is 145 pounds, and its height is 30 inches to the top of cylinder.

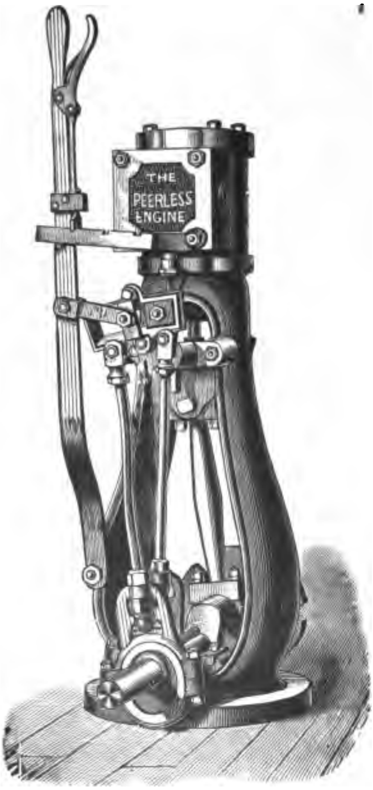

Fig. 71.

THE PEERLESS MARINE ENGINE.

Fig. 74 shows the style of engine which they call the "Peerless" engine. The arrangement of the link is such as to dispense with five or more moving parts, and have the link-block take up its own wear, thus avoiding any rattle at this point.

The Peerless marine engines are among the simplest, most compact and strongest in the market. The crank, connecting-rod, piston-rod and stem, and pins, are made of steel. The iron frame of the engine and the slides, as well as the bearing for the crank-shaft, are made in one piece, so that it is impossible for the working parts of the engine to get out of line or change their relation to each other. The construction of the engine is such that the action of the piston-rod is exactly central, and all overhanging strains are avoided.

The engines that are fitted in launches are what is known as the inverted-cylinder, high-pressure type of engines. They are very simple in construction, not liable to derangement, and with ordinary good usage should run for ten years without any extensive repairs. They have large bearing surfaces, are fitted with steel reversing links, and provision is made for perfect lubrication. All the wearing parts are easy of access, and are adjustable to wear. They are short stroke, quick, powerful little engines, and are very economical of steam.

All the couplings and other parts are keyed on and secured in the most careful manner. The slipping of a shaft in a coupling and consequent stoppage of a small pleasure-boat right in the path of a moving steamer has been known to lead to serious results. Too much care cannot be exercised in regard to these small details, and the integrity of workmanship in launches is worthy the careful consideration of buyers.

While presenting an appearance which, in comparison with a side-wheel steamer, is less graceful, the stern-wheeler* is by no means less efficient in action than its more symmetrical rival, and for small boats this type possesses many advantages. While the weight of one or two persons on the guards of a side-wheel steamer, 60 feet long, and of proportionate beam, will have no perceptible effect in careening the boat, it is found that the boats having a length of 30 feet and under, the effect of such weight on the guards is quite a serious matter. When a stern-wheel is used, however, the weight of the passengers is all within the dimensions of the hull, thus rendering it extremely stable and free from that rolling motion which forms an objectionable feature of small side-wheel steamers. In addition to this the location of the machinery is quite favorable, leaving, as it does, the entire middle portion of the boat available for passengers.

* Fig. 75.

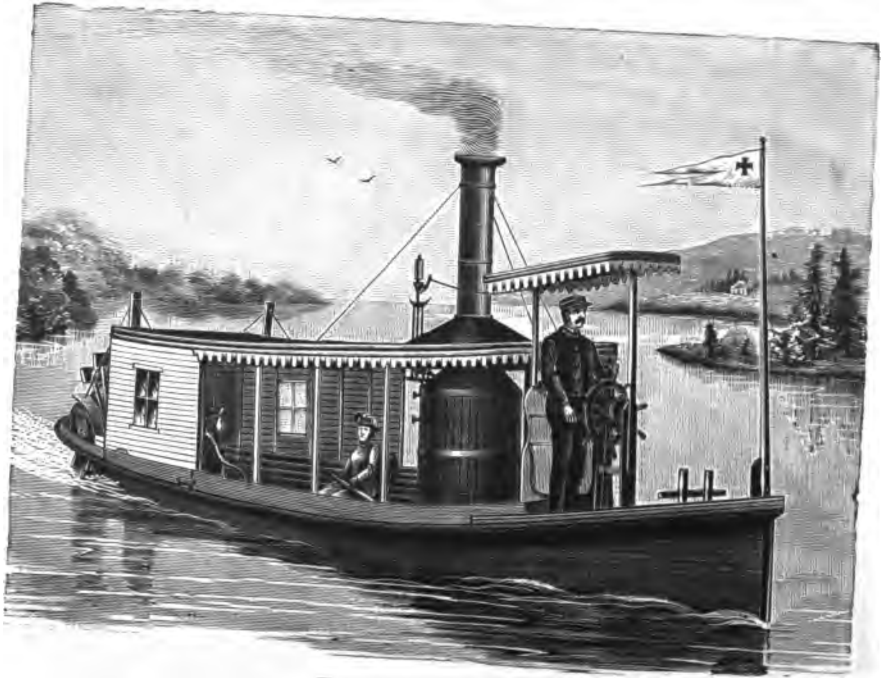

Fig. 75. Page 339.

STERN WHEEL LAUNCHES.

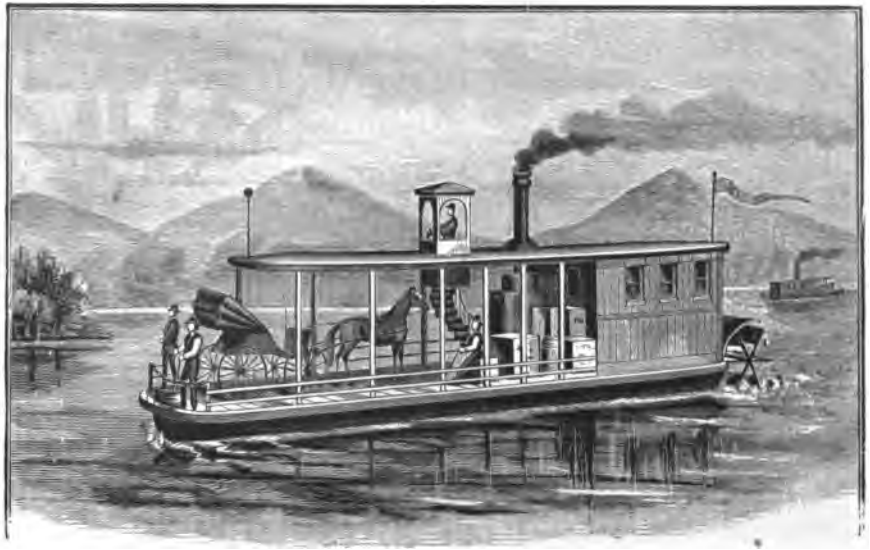

Fig. 76. Page 339.

SMALL PADDLE WHEEL STEAMERS–FERRY BOATS

Fig. 75 represents a stern-wheel launch, showing the arrangement of the machinery, steering-gear and the general design.

These launches are designed with special reference to their use on rivers and lakes where the depth of water prohibits the use of the propeller.

Their light draft, economy of construction, powerful machinery and consequent speed, render them very desirable for shallow waters.

The hulls of these launches should be built of the best material, in a thoroughly workman-like manner, so modelled as to give the greatest speed attainable with a light draft of water. An awning-deck overhead, supported on oak stanchions, should form a protection from the sun and rain, while at the same time being free from the objections that pertain to a close cabin. The steering apparatus should be placed as shown, near the bow, thus enabling one person to make a landing with ease and convenience.

The machinery should consist of a submerged tube vertical boiler, complete in every respect, and complying with the Government rules and regulations relative to marine boilers. The engines, two in number, should be connected direct to the stern-wheel shaft, thus doing away with all intermediate friction, a necessary evil that exists when the stern-wheel is driven by means of gearing. They should be reversed and controlled by a single lever and throttle-valve, in the same manner as all ordinary vertical launch engines.

In order to separate the passengers from the engine and boiler room, cross bulk-heads should be built into the hull, thus completely shutting them out. Between these two bulk-heads and on each side should be arranged the seats.

In operation they will be found exceedingly handy, speedy little boats.

DIMENSIONS. FIG. 75.

Length of hull, feet	33
" boat, " over all	40
Beam, outside, feet	8
Depth, feet	3
Draft, inches	16
Number of persons will carry with comfort	30
Approximate speed, miles per hour	10
Approximate consumption of soft coal, per 10 hours, pounds	400
Actual Horse Power with 70 pounds of steam	10
Engines—two in number—diameter of cylinder, inches	5
" " stroke, inches	20
Vertical boiler, diameter inches	40
" " height, inches	53
Diameter of stern-wheel, feet	7

Although the screw-propeller has almost entirely superseded the paddle-wheel, as a mode of propelling sea-going steamships, the latter still maintains its place in river steamers, and in sea-going steamers of large power and light draft of water.

Inasmuch as the propeller-wheel, to work efficiently, must be submerged, or nearly so, any increase in its size demands an increase in the draft of water, while a paddle-wheel can be increased by making it broader on the face, without increasing the dip of the floats; it follows, then, that where much power is required in combination with a light draft of water, the propeller-wheel can not be used, and recourse must be had to paddle-wheels. For river service, in its two forms, namely, the stern-wheel and side-wheel, it still holds its own.

The magnificent floating palaces that ply upon our western rivers are, almost without exception, driven by one of these two forms; in fact, with the limited draft of water, it would be impossible to use any other known method of propulsion, and attain the speed for which these steamers are famous.

For inland waters, free from the disturbing influences of storms, this method of propulsion has many advantages. The enormous guards on side-wheel steamers, extending over the sides of the boats, give great carrying capacity, either for freight or passengers, while the light draft of water permits a landing at any point along the shore, without injury to the hull.

It is conceded that, for passenger traffic, on rivers the side-wheel steamer is best, and it may be said that there is no other form that combines in so great a degree large carrying capacity with such small dimensions.

The propeller wheels on Willard's launches are made of the best selected cold-blast iron, and are of various forms, according to the use to which the launch is to be put. Where high speed is desired, they furnish a different wheel than for towing purposes. For very small launches a two or three blade wheel will be found desirable, but for launches using wheels 30 inch or larger diameter, the writer would recommend a four-blade wheel.

Fig. 77 shows the different styles of propeller wheels which are furnished with these launches. No. 1 is a towing wheel. No. 2 is a speed wheel; it has a little different shape and pitch than that used for towing. No. 3 is a three-blade speed wheel suitable for a small launch. No. 4 is a two-blade wheel suitable for a small launch, particularly where it is liable to run into grass or weeds, as it is less liable to get fouled than a three or four-blade wheel.

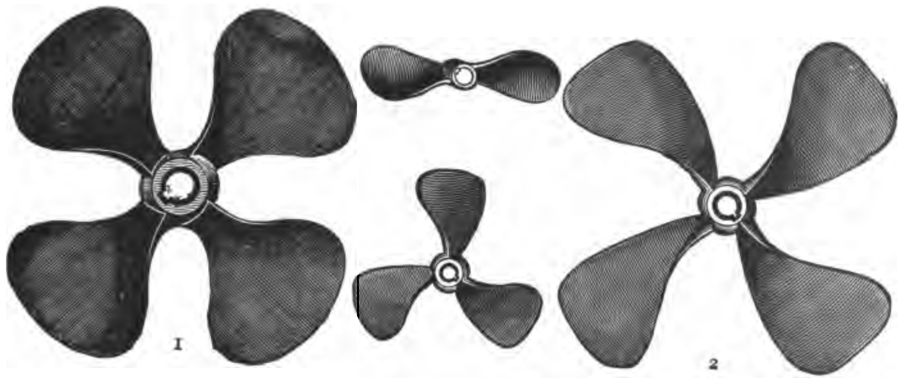

Fig. 77. Page 342.

PROPELLER WHEELS.

STEAM LAUNCHES—SIZES.

NUMBER.	1	2	3	4	5	6	7
Length of Boat, over all	21	25	30	33	40	45	50
Breadth outside .	5	6	7	7½	8	8.6	10
Depth amidship . .	2.4	2.9	3.0	3.0	3.0	3.3	4.6
No. of persons will carry with comfort and safety	9	15	25	30	35	40	60
Approximate speed, miles per hour . .	7	8	9	10	10½	10	11½
Approximate consumption of soft coal per 10 hours, lbs	150	250	350	400	600	750	1,000
Actual Indicated Horse Power, with 100 lbs. steam pressure	3½	8	10	12½	18	24	32½
Engine, Diameter Cylinder	3	4½	5	6	6	7	8
Engine, Length of Stroke	4	5	7	8	8	8	10
No. of Revolutions per minute	300	300	275	275	250	250	200
Vertical Boiler, Diameter	22	26	30	36	40	42	48
Vertical Boiler, Height	36	48	48	50	53	58	72
Propeller Wheel, Diameter	20	24	30	36	36	38	42
Wheel Shaft, Diameter	1¼	1½	1¾	1 15/16	1 15/16	2 3/16	2 7/16

WEIGHT OF ROUND AND SQUARE IRON.

Size in Inches.	Rounds. Weight, lbs.	Squares. Weight, lbs.	Size in Inches.	Rounds. Weight, lbs.	Squares. Weight, lbs.
3/16	.118	.117	1 1/2	5.89	7.50
1/4	.163	.208	1 5/8	7.91	8.80
5/16	.257	.325	1 3/4	8.01	10.20
3/8	.368	.468	1 7/8	9.02	11.71
7/16	.501	.638	2	10.47	13.33
1/2	.654	.835	2 1/8	11.82	15.05
9/16	.833	1.01	2 1/4	13.25	16.87
5/8	1.02	1.30	2 3/8	14.76	18.80
3/4	1.47	1.87	2 1/2	16.36	20.80
7/8	2.00	2.55	2 5/8	18.03	22.96
1	2.61	3.33	2 3/4	19.79	25.20
1 1/8	3.31	4.21	2 7/8	21.63	27.55
1 1/4	4.09	5.20	3	23.56	30.00
1 3/8	5.94	6.30			

WEIGHT PER SQUARE FOOT OF GALVANIZED SHEET IRON.

No. of Wire Gauge.		No. of Wire Gauge.		No. of Wire Gauge.	
No. 29	12 oz.	No. 23	20 oz.	No. 17	36 oz.
" 28	13 "	" 22	22 "	" 16	42 "
" 27	14 "	" 21	24 "	" 15	46 "
" 26	15 "	" 20	27 "	" 14	53 "
" 25	16 "	" 19	30 "	" 13	61 "
" 24	18 "	" 18	35 "	" 12	70 "

WEIGHT OF PLATE IRON PER SQUARE FOOT.

Thickness. in inches.	Lbs.	Thickness. in inches.	Lbs.	Thickness. in inches.	Lbs.
1/16	2.55	9/16	22.66	1 1/8	45.32
1/8	5.03	5/8	25.18	1 1/4	50.35
3/16	7.56	11/16	27.69	1 3/8	55.39
1/4	10.07	3/4	30.21	1 1/2	60.42
5/16	12.59	13/16	32.73	1 5/8	65.46
3/8	15.11	7/8	35.25	1 3/4	70.49
7/16	17.62	15/16	37.76	1 7/8	75.53
1/2	20.14	1	40.28	2	80.56

APPENDIX.

BOILER FELTING OR COVERING.

Magnesium Carbonate as a Non-conductor of Heat.[*]

THE substance referred to in the title is the artificially prepared basic carbonate of magnesia, a compound of the carbonate with the hydroxide. It is the "block-magnesia" of commerce, the *magnesia alba* of the pharmacist. Its composition, though somewhat indefinite, may usually be expressed by the formula: $4MgCO_3, Mg(OH)_2, 5H_2O$; it is found to vary slightly with the details of the method of preparation. This substance has been employed recently with much reported success, as a non-conductor of heat. It is moulded to form coverings suitable for steam-pipes and their fittings, and sectional jackets for boilers and cylinders; it is furnished also in forms suitable for lining refrigerators, walls and roofs of buildings, fire-proof safes, etc. It was with a view of ascertaining the efficiency of this material as a non-conductor, or, rather, its relative efficiency among the various non-conducting materials in use, that the writer made the experiments about to be described.

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[*] A paper read before the American Institute of Mining Engineers at the Scranton Meeting, February, 1887, by Eberhard Luttgen, Ph. B., F. C. S., Ambler, Pa.

A block of magnesium carbonate or *magnesia alba* exhibits some interesting physical qualities. It is a smooth, white, close-grained solid, in outward appearance resembling a block of Paris plaster. It possesses the lightness of cork, the porosity of sponge, and withal a degree of firmness and strength that, in view of its levity, is quite remarkable. To examine more closely the properties of this substance, a number of 1-inch cubes were sawed from the commercial block-carbonate, also some bricks, that is, blocks measuring accurately 2 x 4 x 8 inches, the dimensions of an ordinary brick. The cubes weighed from 2.2 to 2.7 grammes, the bricks weighed 140 to 175 grammes (5 to 6 ounces). The bricks were carefully measured and weighed, and placed in vessels containing distilled water, in which they became gradually submerged, owing to the displacement by water of the air inclosed in the structure of the magnesium carbonate. After twenty-four hours the blocks were removed from the water, dried superficially by contact with filter-paper, and weighed. From the increase in weight, the volume of the water absorbed, and consequently that of the air displaced by it, were obtained. For example: a block which when dry weighed 155 grammes and measured 1048 cubic centimeters, after immersion in water weighed 1138 grammes, an increase of 983 grammes, corresponding to 983 cubic centimeters of water absorbed. We have then the ratio of 983 to 1048, or 93.8 per cent., as the proportion by volume of air-cells in the block of magnesium carbonate. Other blocks, treated in the same way, gave results ranging between 92 and 94.5 per cent. of air-cells. Blocks of magnesium carbonate, upon being saturated with water, undergo no change of size or shape, and upon drying, resume their previous condition.

To ascertain the strength of the blocks, a 1-inch cube was placed between two small boards upon a level surface. Weights were then piled upon the upper board until the cube collapsed. It required 85 to 90 pounds to crush a 1-inch cube. In moulding the magnesium carbonate into coverings for pipes, boilers, etc., the manufacturers incorporate with it a certain proportion of the soft woolly asbestos fiber. This admixture has been found by analysis to amount to from 7 to 8 per cent., and its purpose is to make the coverings stronger and more capable of withstanding rough handling in transportation. This purpose it seems to accomplish, at the same time depriving the coverings only to a slight extent of the lightness and porosity of pure *magnesia alba*.

"Bricks" of this fibrous magnesia weighed about 7 ounces, and were found to contain from 90 to 92 per cent. of air-cells; and the cubes were capable of resisting a greater pressure than were those of the unmixed carbonate.

In investigating the value of a non-conducting material, the first of the various essential qualities to be considered is, naturally, the non-conductivity. The question, what method shall be employed to compare the conductivity of various materials, is somewhat perplexing. The methods that have been pursued have been subjected to much censorious criticism, sometimes just, sometimes unreasonable. While the quantitative results that have been obtained may not, in view of the greater or less inexactness of the methods available, be strictly accurate, there is little doubt of the conformity of their comparative indications respecting the various materials tried. The experiments of Professor Ordway, with the exception of those upon the air-chamber method, were directed towards obtaining quantitative results, towards ascertaining the number of heat-units radiated through a given length of a given pipe-covering in a given time. Whatever doubt may exist, in view of the difficulties of making such determinations, of the accuracy of the quantitative results that were obtained, the experiments furnished a criterion of the comparative values of the materials tried; and for this reason Professor Ordway's report was a very welcome acquisition to the meager literature of non-conducting materials.

In the present case it was desired to subject the carbonate of magnesia covering, together with some of the covering-materials most widely used, to a comparative trial by some simple and expeditious method of testing; the time and facilities for erecting long lines of steam-pipe for the trials not being available. The first method tried was based upon the hypothesis that if equal metallic cylinders be filled with a hot liquid and encased with different non-conducting materials, the rapidity of fall in temperature of the liquid is proportional to the rapidity of radiation through the coverings. The form that suggested itself as the most convenient in which to experiment with non-conducting materials was that in which they are applied to a 2-inch pipe. The cylinders were made of light tin-plate, and were 12 inches long by 2⅜ inches external diameter; the upper end being furnished with a neck for corking. From each of the materials to be tried a cylinder 12 inches in length and fitting the tin closely was cut, and also a disk of the same material for the top and bottom, the upper disk being perforated to admit the neck of the tin cylinder. The cylinders, after being properly encased by the coverings to be tried, were filled with boiling water, and the thermometers inserted through the perforated corks. The fall in temperature of the water was noted at regular intervals, and it was found that repetitions of the experiment under like conditions gave results agreeing closely. Hot glycerine was substituted for the water, and with this a much higher temperature was obtained. It was found, however, that the relative efficiency of the non-conductors was not the same under all conditions; for when the cylinders and their coverings were heated by exposure over night in an oven at a temperature of 270° F., the relative conductivity was reversed with respect to several of the non-conductors. It was accordingly decided to abandon this method of testing for one in which the conditions should approximate more closely to those of actual practice.

The coverings submitted to trial are enumerated below. A sufficient quantity of each for covering 6 feet of 2-inch pipe was purchased from the manufacturers or their agents.

1. *Hair-Felt.*—This was wrapped around the pipe in single thickness and secured by twine wound spirally about it. Around the hair-felt was wrapped a covering of ordinary burlap of single thickness, also fastened by twine wound spirally. The covering was 4½ inches in diameter and weighed 12½ ounces per linear foot.

2. *Sectional Carbonate of Magnesia Covering*, made by the Magnesia Sectional Covering Com-

pany*, Philadelphia.—This consisted of hollow cylinders of the magnesia compound already described, 3 feet in length and bisected longitudinally. The two halves are held together by a jacket of asbestos paper, which forms a hinge. The covering is closed about the pipe and secured by bands of japanned hoop-iron disposed at intervals of 18 inches. This covering was 4½ inches in diameter, and weighed, with the bands, 20¼ ounces per linear foot.

3. *Sectional Carbonate of Magnesia Covering*, the same as No. 2, but having a jacket of cotton duck or light canvas instead of the asbestos-paper jacket. Weight same as No. 2.

4. *Sectional Mineral Wool Covering*, made by the American Steam Covering Company, New York. A hollow envelope of asbestos paper, lined with mineral wool or spun slag, and having an exterior jacket of cotton drill or muslin. The covering is secured to the pipe by twine and bands of muslin, or by pasting down the projecting flap of asbestos paper. Diameter 5¼ inches. Weight 28¾ ounces per linear foot.

5. *Chalmers-Spence Company's Patent Removable Coverings.*—These consist of hollow cylinders of a complex structure. Nearest the pipe is a layer of asbestos paper, next a layer of asbestos fiber, then a layer of sheathing paper, next a layer of hair-felt, and upon this another

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*No. 9, N. Fifth street, Philadelphia, Pa.

layer of sheathing-paper. One side of the hollow cylinder is cut lengthwise, and to apply it to the pipe, the meeting edges are drawn apart, the cylinder sprung upon the pipe, and fastened with double-pointed tacks. The joints are covered by bands of sheathing-paper. Weight, with 6 double-pointed tacks, 28¼ ounces per linear foot. Diameter 4½ inches.

6. *Bradley's Insulated Air Covering*, made by Shields & Brown, New York.—Hollow cylinders 3 feet long, consisting of nineteen layers of stout paper. Nearest the pipe is a sheet of asbestos paper, next two sheets of sheathing-paper, one sheet of asbestos paper, two of sheathing paper, another of asbestos paper, and twelve more of sheathing-paper. The sheathing-paper is alternately smooth and pressed or embossed. The cylinders are placed upon the pipe, the edges drawn together and secured by double-pointed tacks, and the joints sealed with strips of paper. Diameter 4½ inches. Weight, with tacks, 27¼ ounces per linear foot.

7. *Reed's Pipe Covering.*—Paper cylinders composed of about twelve sheets of paper, the inner one of which is asbestos paper. The covering is applied in the same manner as No. 5, by forcing open the edges and springing upon the pipe. Diameter 4¼ inches. Weight, with tacks, 26¾ ounces per linear foot.

8. *Fossil Meal Pipe Covering*, from the Fossil

Meal Company, New York.—A dry mixture of diatomaceous earth with organic fiber (hair, manilla, etc.). Contains probably also some starchy material like flour. It is mixed with water and applied to the pipe with a trowel. Diameter of covering 3¾ inches. Weight 24 ounces per linear foot.

Six feet of each of these coverings was applied in the proper manner to the inclined pipes. The pipes were parallel and placed at such intervals that about three inches space intervened between the coverings. They all received their steam from the main steam pipe, of which they were branches. At each end of the series was an extra pipe, also covered, which served to place the first and last of the coverings under trial in as nearly as possible the same circumstances as the others. The pipes were located in a small engine-room adjacent to the boiler-room; the doors and windows were kept closed, so that there should be no drafts of air during the trials. On the mornings of the days of the trials, the steam was admitted into the apparatus, the stopcocks being closed. At the end of each hour, the water condensed in the pipes was drawn off into graduated cylinders and measured. It was found that during the forenoon the condensations became less each hour, and it was not until 2 or 3 o'clock in the afternoon, when everything was heated to the maximum, that the condensations became uniform. At this point the *conditions of actual practice were established*, and the rate of condensation then existing is that by which we have to judge the efficiency of the different covering materials. The trials were continued several days, and there was no material difference between the results obtained during the different days. The pressure of steam ranged between 65 and 75 pounds to the square inch during the trials, and the temperature of the room was about 70°F. The condensation within the caps, which were well wrapped with cotton-wool, was unimportant with respect to that of the 6 feet of pipe, and was disregarded in calculating the results. The condensation in the differently covered pipes was as follows:

		Grams.		*Grams.*		
No.	1. Hair-felt	208	or	$34\frac{2}{3}$	per foot	per hour.
"	2. Carbonate magnesia.	227	"	$37\frac{5}{6}$	"	"
"	3. Carbonate magnesia.	228	"	38	"	"
"	4. Mineral wool.	231	"	$38\frac{1}{2}$	"	"
"	5. Chalmers-Spence . .	250	"	$41\frac{2}{3}$	"	"
"	6. Shields & Brown . .	255	"	$42\frac{1}{2}$	"	"
"	7. Reed's	270	"	45	"	"
"	8. Fossil meal	345	"	$57\frac{1}{2}$	"	"

The average temperature of the steam being 150.5° C. (302.9° F.), its latent heat was 501.9° C. (935.4° F.). The condensation in No 1 was $34\frac{2}{3}$ grammes per foot per hour. We have, then, as the loss by condensation, $34\frac{2}{3} \times 501.9 \times \frac{1}{1000}$

= 18.39 kilogram-Centigrade heat-units per foot per hour. In the same way we may deduce for the other materials:

No. 2 . .	18.97	kilogram-centigrade	heat-units	per ft. per hr.
" 3 . .	19.07	"	"	"
" 4 . .	19.32	"	"	"
" 5 . .	20.90	"	"	"
" 6 .	21.33	"	"	"
" 7 . .	22.58	"	"	"
" 8 . .	28.86	"	"	"

These results may be reduced to pound-Fahrenheit heat-units by multiplying by $\frac{9}{5} \times 2.205$. This gives for:

No. 1 . .	69.02	pound-Fahrenheit	heat-units	per ft. per hour.
" 2 . .	75.29	"	"	"
" 3 . .	75.68	"	"	"
" 4 . .	76.68	"	"	"
" 5 . .	82.95	"	"	"
" 6 . .	84.65	"	"	"
" 7 . .	89.62	"	'	"
" 8 . .	114.54	"	"	"

From these experiments it appears that, of the materials tried, the hair-felt covering was the most non-conducting. The magnesia covering ranked next, and close upon it came the mineral wool. Following this came the paper coverings, and finally the fossil meal.

All non-conducting materials may be divided, with respect to their composition, into two classes: 1st, Those composed essentially of organic substances, as hair, paper, straw, etc.; and 2d, Those consisting of natural or artificial mineral substances, as asbestos, magnesia, slag-wool, etc.

Of all the organic substances that have been employed as non-conductors, there is probably none that merits recognition as a covering material for hot steam-pipes. This fact has been more saliently thrust upon the attention of steam-users from year to year, by the fires the origin of which is ascribed to the combustibility of this class of pipe-covering materials. The solicitude of the insurance companies has been enlisted in the subject, and they have repeatedly instituted very exhaustive investigations by scientific men, whose reports on these experiments form the major part of the literature of non-conducting materials. Several of these materials, the paper coverings especially, are known to have taken fire spontaneously after being exposed for a long time to the heat of pipes carrying steam under ordinary pressures. And those of the organic materials that do not burst into flame under the conditions prevailing, are, with few exceptions, subject to deterioration by charring, which proceeds with a rapidity and to an extent that depends upon the temperature of the steam used. Aside from the possible danger of conflagrations from these charring and tinder-like substances, the changes which they undergo usually impair their non-conducting qualities. A covering of hair-felt upon a steam-pipe, especially upon a horizontal pipe, will soon become so charred upon the inside, the hair assuming a pulverulent condition, in which it has been compared to snuff, that the covering hangs tangentially to the pipe, forming a channel or flue along the pipe, through which a free circulation of air and escape of heat will take place, if perchance there be any opening in the covering. It has been attempted to prevent the charring of these substances by the interposition of air-spaces and of layers of asbestos paper between the covering material and the pipe, but these attempts have met with only partial success. Asbestos paper is a fair conductor of heat, and while it somewhat retards, does not prevent this charring. Some of these fibrous organic substances, such as hair-felt, cotton-wool, etc., are, in their normal condition, excellent non-conductors of heat; and it is to be regretted that they retain their desirable qualities for only a limited time, and are too often subject to the liability of taking fire.

The mineral non-conducting materials are, as a class, less non-conducting than those composed of organic substances; but they are also less liable to deterioration, and enjoy an immunity from danger of fire. Asbestos, which has been used to some extent for covering steam-pipes, is a tolerably good conductor of heat, and there seems to be little reason for its application as a pipe-covering material. The province of asbestos is rather that of a very convenient incombustible fiber, than that of a non-conductor of heat. It is only when the fiber is in a very soft and downy condition, and full of air, that asbestos is of much value as a non-conductor. But in this respect it never approaches the nature of the structure of slag-wool, which is, at least when fresh, an excellent non-conductor of heat. The mineral wool consists of a mass of extremely fine interlocking fibers, which form multitudes of minute air-chambers, to which is due the non-conductivity of the substance. The fiber is, however, extremely glass-like and brittle, and will not bear much handling. It doubtless merits the reputation which it has, of becoming broken and pulverized by repeated heating and cooling, and by the vibrations and jarring of steam-pipes. The powder then collects at the bottom of the paper-bags, leaving the top of the pipes comparatively unprotected. This reputation the mineral wool shares with the asbestos fiber, but is certainly more entitled to it than the latter substance. Much has been said of the corrosive action of mineral wool upon iron pipes, and Professor Egleston, in a paper read before the American Society of Civil Engineers, has demonstrated that under certain adverse conditions, such a corrosion does take place. But it is probable that mineral wool made from slags that are free from sulphates and sulphides, those of lime particularly, will not be liable to this objection, especially if dampness of the coverings be avoided.

The high degree of excellence of the carbonate of magnesia coverings is due to the myriads of microscopic air-cells contained in the *magnesia alba*, and to its entire freedom from crystalline structure. The paper coverings show a fair degree of non-conductivity, but have the drawback of being short-lived and hazardous. The mineral wool gave better results, but the covering was also much thicker than the paper cylinders. Fossil meal was the poorest of the non-conductors tried, probably because of its too great density. The covering was, however, little over five-eighths of an inch thick, and a greater thickness might have given better results. This thickness (5/8 inch) is that recommended by the makers, and the quantity sent by them to fill an order for covering 6 feet of 2-inch pipe, was no more than sufficient to yield a covering about five-eighths of an inch in depth.

For ease of application and removal, none of the materials tried equaled the magnesia coverings. The sections can be applied to the pipes at a rapid rate by means of the convenient metallic straps. Next come the mineral-wool coverings, and the paper cylinders which are secured by staples, then the hair-felt, and lastly the fossil meal. The application of the latter material requires some practice: a mason was occupied the greater part of a day in plastering the 6-foot pipe experimented with.

In point of appearance the above order also prevails, the magnesia sections being the handsomest, and the mud-like and uneven fossil-meal paste the least sightly of the coverings. The slight difference of radiation between the asbestos-paper-covered and the canvas-covered magnesia sections, was probably more due to an accident in the manufacture, than to the difference in the jacketing material. When absolute indestructibility of the external jacket is less a desideratum than beauty of exterior, the woven jackets are to be preferred. A coat of water-glass or other fire-proof paint would serve to render them uninflammable.

With reference to the economy and cost of non-conducting materials, it may be said that the material which is in the greatest degree non-conducting, incombustible and durable, will prove the most economical, even though its first cost be greater than that of an inferior article. Experiments with naked pipes show that a 2-inch pipe, carrying steam at 60-pounds pressure, will condense 180 grammes per foot per hour. Covered with a good covering like magnesium carbonate, the condensation will be but 38 grammes per foot per hour, a saving of 142 grammes per foot per hour, which is equivalent to 3.13 pounds of steam per day of ten hours for each foot of pipe covered. The covering of 100 feet of pipe, then, will save in a year of 300 ten-hour days, the coal necessary to convert 93,900 pounds of water into steam. One pound of bituminous coal is capable of making about 8.5 pounds of steam, so the saving of coal due to the 100 feet of covering would be 5½ tons per year, which, at $4 per ton, amounts to $22. The real saving will probably amount to more than this estimate in most cases; and it may be said in round terms, that the 100 feet of covering causes each year a saving of its own first cost ($25). Inasmuch as the material pays for itself in a year, and will last indefinitely under ordinary conditions, its advantageousness is beyond question.

From all that precedes, the conclusion drawn regarding the magnesium carbonate coverings is that they are the lightest and most durable of all the mineral non-conductors, and the handsomest and most efficient of all coverings for steam-pipes.

The appended table presents a résumé of the results of the experiments.

Description of Covering.	Diameter of covering in inches.	Weight per foot in ounces av.	Condensed grams per ft. per hour.	Kilog.-Cent. heat units per ft. per hr.	Pound-Fahr. heat units per foot per hour
1. Hair-felt. Wrapped with twine. Burlap jacket.	4½	12½	34.66	17 39	69.02
2. Sectional carbonate magnesia. Asbestos paper jacket. Bands	4½	20¼	37.83	18.97	75.29
3. Sectional carbonate magnesia. Canvas jacket. Bands	4½	20¼	38.00	19.07	75.68
4. Sectional mineral wool. Asbestos paper, mineral wool, muslin	5¼	28¾	38.50	19.32	76.68
5. Chalmers-Spence Co.'s covering. Asbestos, hair-felt, paper	4½	28¼	41.66	20.90	82.95
6. Shields & Brown's covering. Asbestos paper, sheathing paper.	4½	27¼	42.50	21.33	84.65
7. Reed's covering. Asbestos paper, felt paper	4¼	26¾	45.00	22.58	89.62
8. Fossil meal pipe covering. Fossil meal, organic fiber	3¾	24	57.50	28.86	114.54

POSTSCRIPT.

Since the foregoing was written, the author has received for examination a sample of Ainsworth's Plastic Pipe Covering.

This is a smooth, plastic, putty-like material, which appears to consist essentially of clay, paper-pulp and hair. It is applied to the pipe with a trowel, and forms a covering similar in appearance to the Fossil-Meal Covering (No. 8). It was tested in connection with the Magnesia Sectional Covering (No. 3), in the manner already described. The result of several trials was as follows:

Description of covering.	Diameter of covering in inches.	Weight per foot in ounces av.	Condensed grams per ft. per hour.	Kilog.-Cent. heat units per ft. per hr.	Pound-Fahr. heat units per foot per hour.
9. Ainsworth's Plastic Pipe Covering	4½	66	96.6	48.48	192.14
3. Magnesia Sectional Pipe Covering	4½	20¼	38	19.07	75.68

The inefficiency of the Ainsworth covering as a non-conductor of heat is due to its great density and its comparative lack of porosity.

I have also made trials of Plastic Magnesia and the Ainsworth Plastic Covering, the results of which are as follows:

	Diameter.	Condensed gms. per ft. per hr.
Magnesia sectional.	4½	38
Ainsworth's Plastic.	4½	96.6
Magnesia Plastic.	4½	43.2
" "	3½	52.8
" "	5	42.6

THE PREVENTION AND REMOVAL OF SCALE IN STEAM BOILERS.

There is no subject in connection with the use of steam of so much importance as that of the prevention of deterioration of boilers from the injurious effects resulting from an accumulation of scale on the interior surfaces, and from the feed waters containing chemical ingredients, which attack the iron, producing some one of the numerous forms of corrosion, and to which cause, no doubt, are due many disastrous explosions, entailing great loss of life and property: but under more favorable conditions, which, though not culminating disastrously, nevertheless add considerably to the cost in producting the required power. Lord's Compound, a chemical preparation, has been used successfully for many years by more than twenty thousand representative establishments throughout the United States and Canada, and is highly recommended for neutralizing the acids in feed waters, and also for the prevention and removal of scaly deposits, without in any manner injuring the material of the boiler.

This compound is manufactured dry, in the form of a granulated powder, in appearance resembling common brown sugar, and is put up in packages of convenient size—half barrels, barrels, and casks—covering a wide range in weight, of from twenty-five to five hundred pounds, for the convenience of large and small consumers. It readily dissolves in water, and can therefore be applied in a dry state through the man-hole, or in a liquid state by the feed pump, whichever is most convenient.

The quantity for an application will depend upon the nature and amount of water evaporated, the composition of the scale, the construction of the boiler, etc., but for general use a half pound per horse-power per month has been found to give satisfactory results.

It is important in all cases to remove the man- and hand-hole covers at stated intervals, clean out the sediment which is sure to accumulate under all circumstances on the bottom of the boiler, subjecting it to the liability of overheating, and probably rupture; but it is imperatively necessary to be awake to this important duty when using "Lord's Compound," as the scale on the tubes and upper portions of the boiler being detached by it, is in time precipitated to the bottom, increasing the accumulation there and also the danger of overheating, if, as in many cases, the fire is in direct contact with it.

When judiciously applied, this compound will never fail to meet the most sanguine expectations of the purchaser, but being composed of chemicals which are harmless to boiler material, its action may require a longer period than would be needed by many of the dangerous and worthless compounds made of refuse acids, which, while removing the scale, likewise destroy the boiler iron by their corrosive action.

Lord's Compound is probably the only one of its kind that is unanimously endorsed by professional men throughout the length and breadth of this continent, among whom are authors of mechanical books, engineers in charge of works, practical chemists, professional inspectors, and manufacturers having large capital invested in steam boilers. This unanimous endorsement is probably due to the fact that not a single accident has occurred to any boiler using this compound, although they are to be found in every steam-using locality from Canada to Mexico, and from Maine to the Pacific slope. No stronger recommendation of its merits can be given.

This compound is manufactured by Mr. G. W. Lord, 316 Union street, Philadelphia, Pa.

FLOW OF STEAM FROM A GIVEN ORIFICE.

1. Where the steam escapes into a pressure less than half that in the boiler:—

$H = \frac{(p+15) \times 12ak}{7}$; where H = horse-power, p = pressure of steam by guage, a = area of orifice in square inches, and k is a co-efficient = .93 for a short cylindrical pipe, or .63 for a thin opening, or for a safety valve.

2. Where the steam flows into a pressure more than half the pressure in the boiler:—

$H = 3.8\, a\, k \sqrt{(p+15-d)\, d}$; in which d = difference in pressure between the two sides, in pounds per square inch, and a, p and k as above.

3. Where a given horse-power is required to flow through a given opening, to determine the necessary difference in pressure:—

$$d = \frac{p+15}{2} - \sqrt{\frac{(p+15)^2}{4} - \frac{H^2}{14\, a^2\, k}}$$

To reduce either of the above to pounds per hour, multiply H by 30.

For pounds per minute divide H by 2.

PROPERTIES OF SATURATED STEAM.

Pressure.		Temperature in Fahrenheit Degrees.	Volume.		Latent Heat in Fahrenheit Degrees.	Total Heat required to generate 1 lb. of Steam from Water at 32 deg. under constant pressure.
By Steam Gauge.	Total.		Compared with Water.	Cubic ft. of Steam from 1 lb. of Water.		
						In Heat Units.
0	15	212.0	1642	26.36	965.2	1146.1
5	20	228.0	1229	19.72	952.8	1150.9
10	25	240.1	996	15.99	945.3	1154.6
15	30	250.4	838	13.46	937.9	1157.8
20	35	259.3	726	11.65	931.6	1160.5
25	40	267.3	640	10.27	926.0	1162.9
30	45	274.4	572	9.18	920.9	1165.1
35	50	281.0	518	8.31	916.3	1167.1
40	55	287.1	474	7.61	912.0	1169.0
45	60	292.7	437	7.01	908.0	1170.7
50	65	298.0	405	6.49	904.2	1172.3
55	70	302.9	378	6.07	900.8	1173.8
60	75	307.5	353	5.68	897.5	1175.2
65	80	312.0	333	5.35	894.3	1176.5
70	85	316.1	314	5.05	891.4	1177.9
75	90	320.2	298	4.79	888.5	1179.1
80	95	324.1	283	4.55	885.8	1180.3
85	100	327.9	270	4.33	883.1	1181.4
90	105	331.3	257	4.14	880.7	1182.4
95	110	334.6	247	3.97	878.3	1183.5
100	115	338.0	237	3.80	875.9	1184 5
110	125	344.2	219	3.51	871.5	1186.4
120	135	350.1	203	3.27	867.4	1188.2
130	145	355.6	190	3.06	863.5	1189.9
140	155	361.0	179	2.87	859.7	1191.5
150	165	366.0	169	2.71	856.2	1192.9
160	175	370.8	159	2.56	852.9	1194.4
170	185	375.3	151	2.43	849.6	1195.8
180	195	379.7	144	2.31	846.5	1197.2

FLOW OF STEAM THROUGH PIPES.

The quantity of steam flowing through a pipe in a given time under a given "head" or loss of pressure, increases more rapidly than the area of the pipe, or its ratio to the length, directly as the square root of the density, and of the loss of pressure, and inversely as the square root of the length.

The following table gives approximately the weight of steam per minute which will flow from various initial pressures, with a loss of pressure due to friction of one pound per square inch, through different sizes of straight smooth pipes, each having a length of 240 times its own diameter (equal to 20 feet for each inch in diameter). It has been calculated by the formula:

$$W = 303.25\, d^2 \sqrt{\frac{D\,(p_1 - p_2)}{L\left(1 + \frac{3.6}{d}\right)}}$$

In which d=diameter in inches, D=density or weight per cubic foot; p_1 the initial pressure, p_2 the pressure at end of pipe, and L=the length *expressed in diameters.* For sizes of pipe below 6 inch, the flow is calculated from the *actual* areas of "standard" pipe of such nominal diameters.

TABLE OF FLOW OF STEAM THROUGH PIPES.

Initial pressure by gauge. lbs. per sq. in.	Diameter of Pipe in Inches. Length of each = 240 diameters.													
	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
	Weight of Steam per minute in pounds, with 1 pound loss of pressure.													
1 . .	1.16	2.07	5.7	10.27	15.45	25.38	46.85	77.3	115.9	211.4	341.1	502.4	804	1177
10 . .	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7	622.5	996	1458
20 . .	1.70	3.02	8.3	14.94	22.49	36.94	68.20	112.6	168.7	307.8	496.5	731.3	1170	1713
30 . .	1.91	3.40	9.4	16.84	25.35	41.63	76.84	126.9	190.1	346.8	559.5	824.1	1318	1930
40 . .	2.10	3.74	10.3	18.51	27.87	45.77	84.49	139.5	209.0	381.3	615.3	906.0	1450	2122
50 . .	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0	979.5	1567	2294
60 . .	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70 . .	2.57	4.58	12.6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7	1108.5	1774	2596
80 . .	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7	1166.1	1866	2731
90 . .	2.83	5.04	13.9	24.92	37.52	61.62	113.74	187.8	281.4	513.3	828.1	1219.8	1951	2856
100 . .	2.95	5.25	14.5	25.96	39.07	64.18	118.47	195.6	293.1	534.6	862.6	1270.1	2032	2975
120 . .	3.16	5.63	15.5	27.85	41.93	68.87	127.12	209.9	314.5	573.7	925.6	1363.3	2181	3193
150 . .	3.45	6.14	17.0	30.37	45.72	75.09	138.61	228.8	343.0	625.5	1009.2	1486.5	2378	3481

For horse power, multiply the figures in the table by 2. For any other loss of pressure, multiply by the square root of the given loss. For any other length of pipe, *divide 240 by the given length expressed in diameters, and multiply the figures in the table by the square root of this quotient*, which will give the flow for 1 lb. loss of pressure. Conversely, dividing the given length by 240 will give the loss of pressure for the flow given in the table.

The resistance to the steam entering the pipe, when it is not provided with special bell mouth opening, is equal to the friction of 60 diameters additional length, or 25 per cent. additional loss of pressure under the conditions of the table. A globe valve may be estimated as 60, and each elbow as 40 diameters additional length. Thus, a pipe 120 diameters in length, with one globe valve and three elbows, would have 1½ lbs. loss of pressure with the flow given in the table, or deliver $1 \div \sqrt{1\frac{1}{2}} = 81.7$ per cent. of the steam, with one pound loss of pressure. These equivalents, viz :—60 for opening, 60 for each globe valve, and 40 for each elbow must be added in all cases in estimating the length of pipe; thus in the case just given, $120 + 60 + 60 + (3 \times 40) = 300 = 240 \times 1\frac{1}{2}$.

CHIMNEYS.

Chimneys are required for two purposes—1st, to carry off obnoxious gases; 2d, to produce a draught, and so facilitate combustion. The first requires size, the second height.

Each pound of coal burned yields from 13 to 30 pounds of gas, the volume of which varies with the temperature.

The weight of gas carried off by a chimney in a given time depends upon three things—size of chimney, velocity of flow, and density of gas. But as the density decreases directly as the absolute temperature, while the velocity increases, with a given height, nearly as the square root of the temperature, it follows that there is a temperature at which the weight of gas delivered is a maximum. This is about 550° above, the surrounding air. Temperature, however, makes so little difference, that at 550° above, the quantity is *only four per cent.* greater than at 300°. Therefore, height and area are the only elements necessary to consider in an ordinary chimney.

The intensity of draught is, however, independent of the size, and depends upon the difference in weight of the outside and inside columns of air, which varies directly with the product of the height into the difference of temperature. This is usually stated in an equivalent column of water, and may vary from 0 to possibly 2 inches.

To find the maximum draught for any given chimney, the heated column being 612° F., and the external air 62°:—*Multiply the height above grate in feet by .0075, and the product is the draught power in inches of water.*

The intensity of draught required varies with the kind and condition of the fuel, and the thickness of the fires. Wood requires the least, and fine coal or slack the most. To burn anthracite slack to advantage, a draught of 1¼ inch of water is necessary, which can be attained by a well-proportioned chimney 175 ft. high.

A round chimney is better than square, and a straight flue better than tapering, though it may be either larger or smaller at top without detriment.

The effective area of a chimney for a given power, varies inversely as the square root of the height. The actual area, in practice, should be greater, because of retardation of velocity due to friction against the walls. On the basis that this is equal to a layer of air two inches thick over the whole interior surface, and that a commercial horse-power requires the consumption of an average of 5 pounds of coal per hour, we have the following formulæ:

$$E = \frac{0.3\,H}{\sqrt{h}} = A - 0.6\sqrt{A} \quad \ldots \quad 1$$

$$H = 3.33\,E\sqrt{h} \quad \ldots \quad 2$$

$$S = 12\sqrt{E} + 4 \quad \ldots\ldots \quad 3$$

$$D = 13.54\sqrt{E} + 4 \quad \ldots\ldots \quad 4$$

$$h = \left(\frac{0.3\,H}{E}\right)^2 \quad \ldots\ldots \quad 5$$

In which H=horse-power; *h*=height of chimney in feet; E=effective area, and A=actual area in square feet; S=side of square chimney, and D=dia. of round chimney in inches. The following table is calculated by means of these formulæ, by Wm. Kent:

SIZES OF CHIMNEYS WITH APPROPRIATE HORSE-POWER OF BOILERS.

Diam. in inches.	Height of Chimneys, and Commercial Horse-Power.											Side of square inches.	Effective Area, square ft.	Actual Area. square ft.
	50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.			
18	23	25	27									16	0.97	1.77
21	35	38	41									19	1.47	2.41
24	49	54	58	62								22	2.08	3.14
27	65	72	78	83								24	2.78	3.98
30	84	92	100	107	113							27	3.58	4.91
33		115	125	133	141							30	4.48	5.94
36		141	152	163	173	182						32	5.47	7.07
39			183	196	208	219						35	6.57	8.30
42			216	231	245	258	271					38	7.76	9.62
48				311	330	348	365	389				43	10.44	12.57
54					427	449	472	503	551			48	13.51	15.90
60					536	565	593	632	692	748		54	16.98	19.64
66						694	728	776	849	918	981	59	20.83	23.76
72						835	876	934	1023	1105	1181	64	25.08	28.27
78							1038	1107	1212	1310	1400	70	29.73	33.18
84							1214	1294	1418	1531	1637	75	34.76	38.48
90								1496	1639	1770	1893	80	40.19	44.18
96									1876	2027	2167	86	46.01	50.27

The external diameter at the base should be one-tenth the height, unless it be supported by some other structure. The "batter" or taper of a chimney should be from 3/16 to ¼ inch to the foot on each side.

Thickness of brick work: one brick (8 or 9 inches) for 25ft. from the top, increasing ½ brick (4 or 4½ inches) for each 25ft. from the top downwards.

If the inside diameter exceed 5 ft. the top length should be 1½ bricks, and if under 3 ft. it may be ½ brick for ten feet.

STEAM.

A cubic inch of water evaporated under ordinary atmospheric pressure is converted into 1 cubic *foot* of steam (approximately).

Steam at atmospheric pressure flows into a *vacuum* at the rate of about 1550 feet per second, and into the *atmosphere* at the rate of 650 feet per second.

The specific gravity of steam (at atmospheric pressure) is .411 that of air at 34° Fahrenheit, and .0006 that of water at same temperature.

27,222 cubic feet of steam weigh one pound: 13,817 cubic feet of air weigh 1 pound.

Locomotives average a consumption of 3000 gallons of water per 100 miles run.

The best designed boilers, well set, with good draught and skillful firing, will evaporate from 7 to 10 lbs. of water per pound of first-class coal. The average result is from 25 to 60 per cent below this.

In calculating horse-power of Tubular or Flue boilers, consider 15 square feet of heating surface equivalent to one *nominal* horse-power.

One square foot of grate will consume on an average 12 lbs. of coal per hour.

Steam engines, in economy, vary from 20 to 60 lbs. of feed water and from 2 to 7 lbs. of coal per hour per indicated H. P.

Condensing engines require from 20 to 30 gallons of water to condense the steam represented by every gallon of water evaporated—approximately for most engines, we say, from 1 to 1½ gallons per minute per I. H. P. Jet condensers do not require quite as much water for condensing as Surface condensers.

Surface condensers should have about 2 square feet of tube (cooling) surface per horse-power of steam engine. It is absolutely necessary to place air-pumps below condensers to get satisfactory results.

RATIO OF VACUUM TO TEMPERATURE (FAHRENHEIT) OF FEED WATER.

00	inches, Vacuum	212°
11	" "	190°
18	" "	170°
22½	" "	150°
*25	" "	135°
27½	" "	112°
28½	" "	92°
29	" "	72°
29½	" "	52°

WEIGHT AND COMPARATIVE FUEL VALUE OF WOOD.

1 Cord Air-dried Hickory or Hard Maple weighs about 4500 lbs., and is equal to about 2000 lbs. coal.

1 Cord Air-dried White Oak weighs about 3850 lbs., and is equal to about 1715 lbs. coal.

1 Cord Air-dried Beach, Red Oak or Black Oak, weighs about 3250 lbs., and is equal to about 1450 lbs. coal.

1 Cord Air-dried Poplar (whitewood), Chestnut or Elm weighs about 2350 lbs., and is equal to about 1050 lbs. coal.

1 Cord Air-dried Average Pine, weighs about 2000 lbs., and is equal to about 925 lbs. coal.

From the above it is safe to assume that 2¼ lbs. of dry wood is equal to 1 lb. average quality of soft coal, and that the full value of the same *weight* of different woods is very nearly the same—that is, a pound of hickory is worth no more for fuel than a pound of pine, assuming both to be dry. It is important that the wood be dry, as each 10 per cent of water or moisture in wood will detract about 12 per cent from its value as fuel.

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* Usually considered the standard point of efficiency—Condenser and Air Pump being well proportioned.

DUTY OF STEAM ENGINES.

A well-known engineer of high authority gives the following comparative figures showing the economy of high grade steam engines in actual practice:

TYPE OF ENGINE.	Temperature of Feed Water.	Lbs. of Water Evaporated per lb. of Cumberland Coal.	Pounds of Steam per I. H. P. used per hour.	Pounds of Cumberland Coal used per I. H. P. per hour.	Cost of I. H. P. per hour, supposing Coal at $6.00 per ton.
Non-Condensing ..	210°	10.5	29.	2.75	$0.0073
Condensing	100°	9.4	20.	2.12	0.0056
Compound Jacketed	100°	9.4	17.	1.81	0.0045

*** The effect of a good condenser and air pump should be to make available about 10 lbs. more mean effective pressure, with the same terminal pressure; or to give the same mean effective pressure with a correspondingly less terminal pressure. When the load on the engine requires 20 lbs. M. E. P., the condenser does half the work; at 30 lbs., one-third of the work; at 40 lbs., one-fourth, and so on. It is safe to assume that practically the condenser will save from one-fourth to one-third of the fuel, and it can be applied to any engine, cut-off or throttling, where a sufficient supply of water is available.

TABLE OF SPECIFIC GRAVITIES.

	Weight of a cubic inch in pounds.
Copper, cast	.3178

Iron, cast	.263
Iron, wrought	.276
Lead	.4103
Steel	.2827
Gun-metal	.3177

DIVISIONS OF DEGREES OF HEAT.

The thermometer is an instrument for measuring sensible heat. It consists of a glass tube of very fine bore, terminating in a bulb. This bulb is filled with mercury, and the top of the tube is hermetically sealed after all the air has been expelled. The instrument is then put into steam arising from boiling water, and when the barometer stands at 30 inches, a mark is placed on a scale affixed opposite the place the mercury stands at. It is again put in melting ice and the scale again marked. The space between these marks is divided into spaces called degrees. In this country and England it is divided into 180 equal parts, calling freezing point 32°, so that the boiling point is 212, and zero or 0 is 32 below freezing point; and this scale is called Fahrenheit's. On the continent two other scales are in use: the Centigrade, in which the space is divided into 100 equal parts, hence the name; and Reaumur's, in which the space is divided into 80. In both of these scales freezing point is 0 or zero; so that the boiling point of Centigrade is 100° and Reaumur 80°.

AVERAGE BREAKING AND CRUSHING STRAINS OF IRON AND STEEL.

Breaking strain of wrought iron = 23 tons
Crushing strain of wrought iron = 17 tons
Breaking strain of cast iron about 7½ tons
Crushing strain of cast iron = 50 tons . .
Breaking strain of steel bars about 50 tons
Crushing strain of steel bars up to 116 tons

Per square inch of section.

PITTING OF MUD DRUMS.

Mud drums have frequently been known to pit through their close connection to the brick work with which they are covered. When the boiler is filled with cold water, the iron will sweat. This moisture mixing with the lime of the brickwork will, after a length of time, injure the iron. Mud drums are injured on the inside by a similar chemical action. The sediments of lime, etc., deposit there where their action goes on undisturbed by any circulation. To prevent pitting on the inside from this cause, blow down frequently, and on the outside keep the brick off the plates, so that all moisture can pass off.

PROPORTIONS OF VARIOUS COMPOSITIONS IN COMMON USE.

(IN 100 PARTS.)

Babbitt's Metal. . . .	Tin 89, Copper 3.7, Antimony 7.3.
Fine Yellow Brass. . .	Copper 66, Zinc 34.
Gun Metal, Valves, &c.	Copper 90, Tin 10.
White Brass.	Copper 10, Zinc 80, Tin 10.
German Silver	Copper 33.3, Zinc 33.4, Nickel 33.3.
Church Bells	Copper 80, Zinc 5.6, Tin 10.1, Lead 4.3
Gongs	Copper 81.6, Tin 18.4.

Lathe Bushes	**Copper 80, Tin 20.**
Machinery Bearings. .	**Copper 87.5, Tin 12.5.**
Muntz Metal	**Copper 60, Zinc 40.**
Sheathing Metal . . .	**Copper 56, Zinc 44.**

STEAM HEATING.

The advantages of steam heating are set forth by Prof. W. P. Trowbridge.

1. The almost absolute freedom from risk of fire when the boiler is outside of the walls of the building to be heated, and the comparative immunity under all circumstances.

2. When the mode of heating is the indirect system, with box coils and heaters in the basement, almost thorough ventilation may be secured, and it is in fact concomitant with the heating.

3. Whatever may be the distance of the rooms from the source of heat, a simple steam pipe of small diameter conveys the heat. From the indirect heaters underneath the apartments to be heated, a vertical flue to each apartment places the flow of the low heated currents of the air under the absolute control of the occupants of the apartment. Uniformity of temperature, with certainity of control, may be thus secured.

4. Proper hygrometric conditions of the air are better attained. As the system supplies large volumes of air heated only slightly above the external temperature, there is but little change in the relative degree of moisture of the air as it passes through the apparatus.

5. No injurious gases can pass from the furnace into the air flues.

6. When the method of heating is by direct radiation in the rooms, the advantages of steadiness and control of temperature, sufficient moisture and good ventilation, are not always secured; but this is rather the fault of design, since all these requirements are quite within reach of ordinary contrivances.

7. One of the conspicuous advantages of steam heating is that the most extensive buildings, whole blocks, and even large districts of a city, may be heated from one source, the steam at the same time furnishing power where needed for ventilation or other purposes, and being immediately available also for extinguishing fires, either directly or through force pumps.

STEAM COAL.

Steam coal being, as everybody knows, unquestionably the most important and largest expense in the manufacture of steam, is deserving a most careful investigation by engineers and owners, who, unlike chemists and college professors, consider the subject wholly in a practical way as relating to the coal bills of their establishments.

Useful knowledge of every-day economy of coal is seldom gained by "tests" conducted by experts, for several reasons so plain that they will not require explanation. 1st. The cost of the fuel used in "tests," whatever may be stated, is too high, average or "every-day" coal not being used. The experiments are made with picked men and picked fuel, for brief periods, with everything at its best, and the results obtained, if looked for in the ordinary run of business, will be disappointment in the results of the wholesale order. 2d. Men working as firemen twelve or fourteen hours per day in the hot furnace rooms, cannot be expected, with the ordinary appliances, to watch where every lump of coal falls when feeding the furnaces, nor to clean the grates any oftener than they are compelled to do. 3d. Moreover, too many employers favor the low wages plan, and for an apparent saving of a few dollars per month, waste many times the amount in their furnace doors, and render their establishments most disagreeable to the neighbors by a free distribution of unconsumed carbon, or what is commonly called *soot*, and of which most people have no appreciation. 4th. Little or no encouragement is given for careful or economical firing, as a rule. The fireman who oftentimes wastes as much as his entire wages, secures the same pay as the man working alongside of him who saves it all. It may be remarked that this is "not business," but many are the concerns who run their steam plants upon this system. Careful handling of coal in firing pays better than any other thing about a steam plant, and it is the wisest economy to secure good and careful men to do it.

As is well understood, the conditions or circumstances attending the combustion of coal for steam purposes, embrace a wide range. A very few establishments work under conditions that admit of a high attainment of economy by having a fixed performance of duty, and their plants well proportioned to the regular work; but by far the largest number have a fluctuating demand for steam, and in that respect are largely at a disadvantage. Many furnaces are badly constructed, others suffer from an insufficiency of draft, and in many cases there seems to be no end of complications detrimental to best results.

These practical difficulties and uncertanties, which are well known to every experienced engineer, render any investigation worthy of the name slow and laborious. It has taken considerable time and research to arrive at the conclusion, though differing from the preponderance of hearsay or guess-work evidence, that now at least "*the highest priced coal is not the cheapest for steam production*," and that in fact the reverse is undoubtedly true, especially in the Western country. Late improvements in the construction of grate-bars have undoubtedly added largely to the value of Western soft coals. The great difficulty in former times of ridding the furnaces of the incombustible part of these very valuable coals has now been removed by improvements, and there is no doubt but that a large number of extensive establishments in the west are now, and for some time past have been, obtaining the same duty from the Illinois bituminous coals that they in former years obtained from the high-priced Eastern coals.

AMERICAN COALS.

Coal. State. Kind of Coal.	Per cent. of Ash.	Theoretical Value.		Coal. State. Kind of Coal.	Per cent. of Ash.	Theoretical Value.	
		In Heat Units.	In Pounds of Water evap.			In Heat Units.	In Pounds of Water evap.
Penn . Anthracite	3.49	14,199	14.70	Ill . . Bureau Co.	5.20	13,025	13.48
“ . “	6.13	13,535	14.01	“ . . Mercer Co.	5.60	13,123	13.58
“ . “	2.90	14,221	14.72	“ . . . Montauk	5.50	12,659	13.10
“ . . . Cannel	15.02	13,143	13.60	Ind . . . . Block	2.50	13,588	14.38
“ Connellsville	6.50	13,368	13.84	“ . . . Caking	5.66	14,146	14.64
“ Semi-bi’nous	10.77	13,155	13.62	“ . . . Cannel	6.00	13,097	13.56
“ Stone’s Gas	5.00	14,021	14.51	Md. Cumberland	13.98	12,226	12.65
“Youghiogheny	5.60	14,265	14.76	Ark . . . Lignite	5.00	9,215	9.54
“ . . . Brown	9.50	12,324	12.75	Col. . . . “	9.25	13,562	14.04
Kentucky.Caking	2.75	14,391	14.89	“ . . . “	4.50	13,866	14.35
“ . Cannel	2.00	15,198	16.76	Texas . . “	4.50	12,962	13.41
“ . “	14.80	13,360	13.84	Wash. Ter. “	3.40	11,551	11.96
“ . Lignite	7.00	9,326	9.65	Penn. Petroleum		20,746	21.47

“Slack,” or the screenings from coal, when properly mixed—anthracite and bituminous—and burned by means of a blower on a grate adapted to it, is nearly equal in value of combustible to coal, but its percentage of refuse is greater.

TEMPERATURE OF FIRE.

By reference to the table of fuels, it will be seen that the temperature of the fire is nearly the same for all kinds of combustibles, under similar conditions. If the temperature is known, the conditions of combustion may be inferred. The following table, from M. Pouillet, will enable the temperature to be judged by the appearance of the fire:

Appearance.	Temp. Fah.	Appearance.	Temp. Fah.
Red, just visible . .	977°	Orange, deep	2010°
" dull	1290	" clear	2190
" Cherry, dull . . .	1470	White heat	2370
" " full . . .	1650	" bright	2550
" " clear . .	1830	" dazzling	2730

To determine temperature by fusion of metals, etc.

Substance.	Temp. Fah.	Metal.	Temp. Fah.	Metal.	Temp. Fah.
Tallow . . .	92°	Bismuth . .	518°	Silver, pure	1830°
Spermaceti .	120	Lead	630	Gold Coin	2156
Wax, white .	154	Zinc	793	Iron Cast, medm .	2010
Sulphur . . .	239	Antimony . .	810	Steel	2550
Tin	455	Brass	1650	Wrought Iron . . .	2910

MACHINE DESIGNING.

Machine designing naturally divides itself into sections, and these we propose to consider separately.

Principle.—This is the first point to be observed—the designer must not design a machine or piece of work that is absurd in principle. No degree of painstaking will overcome this vital defect. We may define principle as taking care that none of Nature's laws are violated. No matter how well the design or drawing is made, if it is wrong in principle it is useless. For instance, no drawing of a machine, or combination of mechanism, for producing perpetual motion would be of any use. We once knew a water-wheel designed to work a pump, and the pump to work the water-wheel; the water-wheel had a crank on the shaft, the crank was to work the plunger-rod, and the pump was to pump water into a trough above the wheel, this trough conveying the water to the buckets of the water-wheel, and thence the whole cycle was to begin again. The drawing looked very well on paper, and was correct so far as the various parts were concerned, but the principle being wrong, no amount of designing and drawing could overcome the defect. The error was pointed out to the designer, but he would not be convinced; he constructed a model, spent some money in experimenting, but all to no effect. The result was that he became a sadder but a wiser man, for he had discovered that perpetual motion was a breach of one of Nature's laws. Another important point bearing upon the principle of designing machinery is that the action should be continuous, if it does not perform the operation at once. In a steam riveter the stroke of the ram makes the rivet head at one operation; a steam hammer does not accomplish its work at one blow, but the effort is continuous; it repeats the blow until the work is accomplished; there is no undoing of work. We knew an instance in which a machine was designed for bending bars, the preliminary part being to straighten them before the curve was put in; when the bar came out of the machine it was curved, but if not to the required curve, the operation could not be repeated without first straightening the bar again, before any additional curving could be put in. This was a case in which the action was not continuous; the first part of the machine undid what had been done by the latter part. Hence the machine, when made, turned out a complete failure, for want of being designed in accordance with correct principle.

Position.—As soon as we are satisfied that the principle is correct, the important point of position comes to the front. To some extent it may be included in principle, which we have defined to be "taking care that no natural law is violated," for the foundation of position is due to the natural law or axiom that no two things can be in the same place at the same time. This is more important to bear in mind when designing moving machinery. A drawing only shows one position of a moving part; hence, unless there is great care on the part of the draughtsman in following out the motion of each part to its full extent, he is liable to have some parts foul one another when they have moved into a new position not shown on the drawing. To some extent this may be avoided by making diagrams showing successive phases of the moving parts, but this involves a great deal of work, and is often neglected. It is desirable, when possible, to show the parts in the worst possible position in relation to each other. Very frequently the extreme phase of one part does not coincide with the extreme phase of another, as in the steam engine the valve is not at full stroke at the same time as the piston. To get over this difficulty, it is very customary to make a working drawing disjointed—that is to say, the various parts are drawn as if they were all in their worst position at once, it being understood that this is only a convention, in order that the various parts may be in their proper position in all phases of the machine, for if everything clears in its worst positions it naturally clears in every other position. In spite of all drawings, however, errors will arise, and the constant care of the draughtsman should always be directed to the point technically called clearance.

It should also be borne in mind that not only have the parts to work when they are together, but that they have to be put together; there should be no collars, for instance, on a shaft to prevent a wheel being slipped endways. Putting work together is naturally an extension of position in relation to moving parts, for the parts are moved, although but seldom, viz., when the machine is put together or pulled to pieces. Whilst considering this point it is well to mention bolts and nuts; the designer should take care that they are arranged where they can be put in, and where the nuts can be put on and taken off. This is a small point, but is one frequently overlooked by draughtsmen, who often show work as if the bolts and other minor parts had grown, it being impossible to get the work together as shown on their drawing.

Strength. — After principle and position, strength may be considered the next factor. This is a matter primarily of calculation, and secondly of precedent, it being well known that if a part of a machine was strong enough in some other machine, or under similar circumstances, it is strong enough for the case under consideration. The strength of materials, and parts of machines, would require a volume to itself, and it is not our intention to dwell upon it. It is important that the designer should be a good calculator, well up in theory, and also that he should have had a good deal of practice, or, in other words, a good store of precedents, which is only theory embodied in practice. Particular care should be taken that no sudden changes occur in the dimensions of bars, axles, and other parts, and also that there should be no internal sharp corners to any of the castings. These are great pitfalls to the unwary, and care should be taken in avoiding them.

Proportions.—The vital points of principle, position and strength being supposed to be settled, there remain what may be considered the luxurious points of pleasing proportions and of ornamentation. Elegance of proportion is more a matter of fancy than anything else, and can only be acquired by practice. There are generally various ways of drawing and arranging the parts of a machine without violating any of the vital points we have already dwelt upon. The draughtsman will do well to think to himself as he goes along: Does this look well? Will it look better in any other way?

Ornamentation in machinery is generally the addition of a few mouldings, scrolls, and such like, which, as in proportion, is a matter of taste or feeling. Care should be taken to avoid contradiction in ornament; there should be a similarity of style about the mouldings, the scrolls, and the shapes of the subordinate parts of the machinery.

It must be understood that all the points into which this article is divided must be considered by the designer simultaneously; they cannot be separated when designing, as we have done here for the convenience of treatment, and in relation to the importance of the divisions.

FOAMING IN BOILERS.

The causes are dirty water; trying to evaporate more water than the size and construction of the boiler is intended for; taking the steam too low down; insufficient steam room; imperfect construction of boiler, and too small a steam pipe.

Take a kettle of dirty water and place it on a fire and allow it to boil and watch it foam, and it will be the same in a boiler.

Too little attention is paid to boilers with regard to their evaporating power. Where the boiler is large enough for the water to circulate, and there is surface enough to give off the steam, foaming never occurs. As the particles of steam have to escape to the surface of the water in the boiler, unless that is in proportion to the amount of steam to be generated, it will be delivered with such violence that the water will be mixed with it and cause what is called foaming.

A high pressure insures tranquillity at the surface, and the steam itself being more dense it comes away in a more compact form, and the ebullition at the surface is no greater than at a lower pressure. When a boiler foams we close the throttle to check the flow, and that keeps up the pressure and lessens the sudden delivery.

Too many flues in a boiler obstruct the passage of the steam from the lower part of the boiler on its was to the surface; this is a fault in construction, but nearly all foaming arises from dirty water, or from trying to evaporate too much water without heating surface or steam room enough. Usually, when first put in, a boiler and engine are large enough; but as business increases more machinery is added, until the power required is greater than can be furnished by the engine, more pressure has to be carried and the number of revolutions increased, consequently the evaporating power of the boiler is forced beyond its ability, the steam being drawn off so rapidly that a large portion of water is drawn with it—so much that it would astonish any engineer if he had a testing apparatus attached to the steam pipe.

For the remedy of foul water there are numerous contrivances to prevent it from entering the boiler, which is a far better way than trying to extract the sediment after it is there—though there are many ingenious methods for doing that also. Faulty construction, or lack of capacity, the engineer cannot help, but he soon learns how to run the boiler to get the best possible results from it.

Every intelligent engineer has observed that his engine has an individuality not possessed by any other he ever ran, and nothing but personal acquaintance can get the best work out of it; so it is with the boiler.

The steam pipe may be carried through the flange six inches into the dome, which would prevent the water from entering the pipes by following the sides of the dome as it does.

For violent ebullition a plate hung over the hole where the steam enters the dome from the boiler is a good thing, and prevents a rush of water by breaking it when the throttle is opened suddenly.

Clean water, plenty of surface, plenty of steam room, large steam pipes, boilers large enough to generate steam without forcing the fires, are all that is required to prevent foaming. A surface blow-off is a grand thing, and helps a foaming boiler, and would be a good thing on every boiler, as you can then skim it as you would an open kettle.

STEAM GAUGES.

Steam-gauges indicate the pressure of steam above the atmosphere, the total pressure being measured from a perfect vacuum, which will add 14 7-10 lbs. on the average to the pressure shown on the steam-gauge.

BLOWING OFF UNDER PRESSURE.

A boiler can be seriously impaired by blowing it down under a high pressure, and with hot brick-work. The heat from the latter will granulate the iron and reduce its tensile strength. A boiler should not be blown right down under a higher pressure than twenty pounds and not less than four hours after the fire has been drawn.

When a boiler is exposed to cold air, especially in the winter, it is advisable that the damper be closed and the doors thrown open, or *vice-versa.* If both are left open the strong draught of cold air will cool off the flues faster than the shell; which abuse if kept up would reduce the length of the life of the boiler.

POLISHING AND FINISHING METALS.

Take, for instance, a surface of steel as an example—the square stem of a drilling instrument will form a very good subject. After it is roughed out, and the work all done, it must be draw-filed, and this must be done with a superfine American file, and the lines must be kept quite straight, otherwise it will require so much emery-paper that the edges will lose the sharp angles which are the beauty of the work. Any ordinary workman can rub away with emery-paper, but in so doing he may spoil the appearance of a piece of good work, and that without knowing it. To avoid this, the smoother and better it is filed the less paper will it require. To get the beautiful finish we see on the best work, a piece of flour emery-paper, well worn, and a little oil upon it, will be found the best thing to use, and when this has been well worked, to get the high polish, a piece of wood flat upon the surface, with some fine crocus, will bring it up to this state; and if any deep scratches be there, you will at once observe them; to remove them, in all probability, it will have to be filed over again. Now, to avoid all this loss of time, great care must be taken that the scratches are removed before any attempt is made to polish. Having finished the work so far, many prefer to see it left straight; others, again, like to see it in some way ornamented. Now, there are several ways of doing this. First, then, to cross the surface. This is done by folding a piece of emery-paper tightly round a file, but the process is not the merely pushing it across the work and making a mark, but it requires some practice to produce a good pattern, and the wrist must take a kind of circular action, and by doing this each line becomes, so to speak, connected, and makes a much better finish than a series of lines only. Another process of finishing steel is to curl all over the surface with a piece of oil-stone that will cut. This is a most difficult thing to obtain, as very few stones will cut steel to leave the bright marks necessary to give it the appearance desired. When a piece of this is once obtained it is really a prize, and if it wears away it may be inserted as far as possible into a wooden handle. To use the stone when it is once obtained is the next thing. This is done by holding it firmly in the hand and moving it about in all directions, like curling brass. There is no stated number or size of the curl, but this is quite a matter of taste, and must be left to the operator. Another way of finishing iron and steel is with the scraper, which is used with both hands, and the work must be scraped in various directions, but with regularity. Large surfaces are sometimes done in this way. Lathe beds are at times done so, but we think this is somewhat out of character, as the fact of continually drawing the poppet head up and down the bed produces a series of lines which look most unsightly. Regarding all this, it is all a matter of taste, and the style of finish must be left to the operator.

WORKING STEEL FOR TOOLS.

In working steel for tools, great care should be taken to hammer all sides alike, for if one side is hammered more than another it will cause it to spring in hardening. Again, steel, when being hammered, should be heated as hot as it will stand, until finishing, and should then be hammered until almost black hot, for the reason that it sets the grain finer, and gives the tool a better edge. The reason for heating the steel so hot while hammering is simply because it makes the steel tougher when hardened, and softer when annealed, while, if it were worked at a low red heat, the continued percussive shocks of the hammer would so harden it as to make it almost impossible to anneal it, and at the same time render it brittle when hardened.

TO HARDEN AND TEMPER CAST STEEL.

For saws and springs in general the following is an excellent liquid: Spermaceti oil, 20 gals.; beef suet *rendered*, 20 lbs.; neat's-foot oil, 1 gal.· pitch, 1 lb.; black resin, 3 lbs. The last two articles must be previously melted together, and then added to the other ingredients, when the whole must be heated in a proper iron vessel, with a close cover fitted to it, until all moisture is evaporated; the composition will take fire on a flaming body being presented to its surface.

TEMPERING POINTS OF TOOLS.

After being tempered the volume of the tool is slightly increased, and consequently its specific gravity is decreased. As the expansion or increase of volume is so very slight, it is quite immaterial which is plunged into the liquid first; however, every moment the edge is kept out it is cooling, and the tempering may be rendered defective thereby. Mercury tempers the hardest, then water, then salt and water, then oil of various kinds—as whale oil. As oil cools the metal more slowly, it is not tempered so hard, but the tenacity is increased.

RULE FOR CALCULATING SPEED OF PULLEYS.

Example 1. To find the size of driving-pulley, multiply the diameter of the driven by the number of revolutions it should make, and divide the product by the revolutions of the driver; the quotient will be the size of the driver.

Example 2. Diameter and revolutions of driver being given, to find the diameter of the driven that shall make a given number of revolutions, multiply the diameter of the driver by the number of revolutions, and divide the product by the number of revolutions of the driven; the quotient will be the size of the driven.

Example 3. To find number of revolutions of the driven pulley, multiply the diameter of the driver by its number of revolutions, and divide by the diameter of the driven. The quotient will be the number of revolutions of the driven.

The following examples will assist in determining diameters and speeds of pulleys:

A 30-inch pulley making 180 revolutions per minute, drives a counter-shaft with a 12-inch pulley. What is the speed of the latter? Ans. $180 \times 30 \div 12 = 450$ revolutions per minute.

A countershaft is to make 450 revolutions per minute, driven by a 30-inch pulley making 180 revolutions per minute. What will be the diameter of the countershaft pulley? Ans. $180 \times 30 \div 450 = 12$ inches.

What will be the diameter of a pulley making 180 revolutions per mintue, to drive a 12-inch pulley 450 revolutions per minute? Ans. $450 \times 12 \div 180 = 30$ inches.

In calculating toothed gears, substitute the numbers of teeth for the diameters, as above.

TO MEASURE BELTING IN THE ROLL.

A simple method of measuring belting in the roll, and which is said to be very closely correct, is as follows: The sum of the diameters of the roll and the eye in inches, multiplied by the number of turns made by the belt, and this product multiplied by the decimal .1309, will be the length of the belt in feet.

DIRECTIONS FOR BRAZING BAND SAW BLADES.

File the two ends of the blade to a featheredge, so that they will lap from one-half inch to one inch, according to the width of the blade. The ends should then be cleaned with *prepared soldering fluid.* Then clamp the two ends in a brazing frame, and see that they are lapped so that the teeth will match, taking great care not to handle the filed portion of the lap, and see that no grease or dirt gets into the joint. Put a small piece of silver solder, rolled thin, in the joint, and clamp the joint with a pair of brazing tongs heated to a cherry red, or a little hotter for wide saws. Keep the tongs clamped firmly on the joint until nearly black. Do not put any water on the braze or it will make it brittle and liable to break easily. When the braze is perfectly cool, file it to an even thickness with the rest of the blade and file the teeth.

In hanging up band saws be careful to hang them so that a short bend will not come at the joint.

DIRECTIONS FOR PUTTING RUBBERS ON BAND SAW WHEELS.

First thoroughly clean the wheel, so that the face is perfectly free from dirt and grease. A particle of grease will spoil the job. When the wheel is perfectly clean and dry, apply with a brush a *coating of special cement.* The inside of the rubber band should then receive a coat of cement also. When this is done place the wheel in a horizontal position and then have three or four men, whose hands must be free from dirt and grease, stretch the rubber evenly on the wheel. The wheel should be then put in a warm dry place for five or six hours. If convenient it can be put in a steam box where it will get dry heat over 200 degrees. Never roll the wheel on the floor and *do not lift* up the *edge of the band* to see if it sticks tight, as this will invariably leave a place where the air can get in, and as soon as the wheel is in motion it will cause the band to loosen. After the wheel is put on the machine, see if it runs perfectly true, and if there are any high places in the band reduce them with a piece of sand or emery paper.

TRANSMITTING EFFICIENCY OF TURNED IRON SHAFTING AT DIFFERENT SPEEDS.

As Prime Mover or Head Shaft carrying Main Driving Pulley or Gear, well supported by bearings.

Diam. of Shaft.	Number of Revolutions per Minute.										
	60	80	100	125	150	175	200	225	250	275	300
Inches.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
1¾	2.6	3.4	4.3	5.4	6.4	7.5	8.6	9.7	10.7	11.8	12.9
2	3.8	5.1	6.4	8	9.6	11.2	12.8	14.4	16	17.6	19.2
2¼	5.4	7.3	8.1	10	12	14	16	18	20	22	24
2½	7.5	10	12.5	15	18	22	25	28	31	34	37
2¾	10	13	16	20	24	28	32	36	40	44	48
3	13	17	20	25	30	35	40	45	50	55	60
3¼	16	22	27	34	40	47	54	61	67	74	81
3½	20	27	34	42	51	59	68	76	85	93	102
3¾	25	33	42	52	63	73	84	94	105	115	126
4	30	41	51	64	76	89	102	115	127	140	153
4½	43	58	72	90	108	126	144	162	180	198	216
5	60	80	100	125	150	175	200	225	250	275	300
5½	80	106	133	166	199	233	266	299	333	366	400

As Second Movers or Line Shafting. Bearings 8 feet apart.

Diam. of Shaft.	Number of Revolutions per Minute.										
	100	125	150	175	200	225	250	275	300	325	350
Inches.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
1¾	6	7.4	8.9	10.4	11.9	13.4	14.9	16.4	17.9	19.4	20.9
1⅞	7.3	9.1	10.9	12.7	14.5	16.3	18.2	20	21.8	23.6	25.4
2	8.9	11.1	13.3	15.5	17.7	20	22.2	24.4	26.6	28.8	31
2⅛	10.6	13.2	15.9	18.5	21.2	23.8	26.5	29.1	31.8	34.4	37
2¼	12.6	15.8	19	22	25	28	31	35	38	41	44
2⅜	15	18	22	26	29	33	37	41	44	48	52
2½	17	21	26	30	34	39	43	47	52	56	60
2¾	23	29	34	40	46	52	58	64	69	75	81
3	30	37	45	52	60	67	75	82	90	97	105
3¼	38	47	57	66	76	85	95	104	114	123	133
3½	47	59	71	83	95	107	119	131	143	155	167
3¾	58	73	88	102	117	132	146	162	176	190	205
4	71	89	107	125	142	160	178	196	213	231	249

For Simply Transmitting Power.

Diam. of Shaft.	Number of Revolutions per Minute.										
	100	125	150	175	200	233	267	300	333	367	400
Inches.	H.P.	H.F.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
1½	6.7	8.4	10.1	11.8	13.5	15.7	17.9	20.3	22.5	24.8	27
1⅝	8.6	10.7	12.8	15	17.1	20	22.8	25.8	28.6	31.5	34.3
1¾	10.7	13.4	16	18.7	21.5	25	28	32	36	39	43
1⅞	13.2	16.5	19.7	23	26.4	31	35	39	44	48	52
2	16	20	24	28	32	37	42	48	53	58	64
2⅛	19	24	29	33	38	44	51	57	63	70	76
2¼	22	28	34	39	45	52	60	68	75	83	90
2⅜	27	33	40	47	53	62	70	79	88	96	105
2½	3	39	47	54	62	73	83	93	104	114	125
2¾	41	52	62	73	83	97	111	125	139	153	167
3	54	67	81	94	108	126	144	162	180	198	216
3¼	68	86	103	120	137	160	182	205	228	250	273
3½	85	107	128	150	171	200	228	257	285	313	342

WROUGHT IRON PIPE.

Size.	Weight, lbs. per foot.
⅛	.24
¼	.42
⅜	.56
½	.85
¾	1.12
1	1.67
1¼	2.25
1½	2.69
2	3.66
2½	5.77
3	7.54
3½	9.05
4	10.72
4½	12.49
5	14.56
6	18.77
7	23.41
8	28.35
9	34.07
10	40.64
12	54.65

TABLE OF HORSE POWER WHICH MAY BE TRANSMITTED BY OPEN SINGLE BELTS TO PULLEYS RUNNING 100 REVOLUTIONS PER MINUTE, THE DIAMETERS OF THE DRIVING AND DRIVEN PULLEYS BEING EQUAL.

The Horse Power of Double Belts is 10-7 of that given in the Table.

Diameter Pulley.	Width of Belt in Inches.							
	2	2½	3	3½	4	4½	5	6
In.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.
6	.44	.54	.65	.76	.87	.98	1.09	1.31
6½	.47	.59	.71	.83	.95	1.07	1.19	1.42
7	.51	.64	.76	.89	1.01	1.14	1.27	1.53
7½	.55	.68	.82	.95	1.09	1.23	1.36	1.64
8	.58	.73	.87	1.02	1.16	1.31	1.45	1.75
8½	.62	.77	.93	1.08	1.24	1.39	1.55	1.86
9	.65	.82	.98	1.15	1.31	1.48	1.64	1.97
9½	.69	.86	1.04	1.21	1.39	1.56	1.74	2.08
10	.73	.91	1.09	1.27	1.45	1.63	1.81	2.18
11	.8	1.	1.2	1.4	1.6	1.8	2.	2.4
12	.87	1.09	1.31	1.53	1.75	1.97	2.18	2.62
13	.95	1.18	1.42	1.65	1.89	2.12	2.36	2.83
14	1.02	1.27	1.52	1.77	2.02	2.27	2.53	3.05
15	1.09	1.36	1.64	1.91	2.19	2.46	2.73	3.29
16	1.16	1.45	1.74	2.03	2.32	2.61	2.91	3.48
17	1.24	1.55	1.85	2.16	2.47	2.78	3.09	3.70
18	1.31	1.64	1.96	2.29	2.62	2.95	3.27	3.92
19	1.39	1.73	2.07	2.42	2.76	3.11	3.45	4.14
20	1.45	1.82	2.18	2.55	2.91	3.27	3.64	4.36
21	1.52	1.91	2.29	2.67	3.05	3.44	3.82	4.58
22	1.66	2.	2.4	2.8	3.2	3.6	4.	4.8
23	1.67	2.09	2.51	2.93	3.35	3.75	4.18	5.02

Diameter Pulley.	Width of Belt in Inches.							
	4	5	6	8	10	12	14	16
In.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.
24	3.5	4.4	5.2	7.	8.7	10.5	12.2	14.
25	3.6	4.5	5.5	7.3	9.1	10.9	12.7	14.5
26	3.8	4.7	5.7	7.6	9.5	11.3	13.2	15.1
27	3.9	4.9	5.9	7.8	9.8	11.8	13.7	15.6
28	4.1	5.1	6.1	8.1	10.2	12.2	14.3	16.3
29	4.2	5.3	6.3	8.4	10.5	12.6	14.8	16.9
30	4.4	5.4	6.6	8.7	10.9	13.1	15.3	17.4
31	4.5	5.6	6.8	9.	11.3	13.5	15.8	18.
32	4.7	5.8	7.	9.3	11.6	14.	16.3	18.6
33	4.8	6.	7.2	9.6	12.	14.4	16.8	19.2
34	4.9	6.2	7.4	9.9	12.4	14.8	17.3	19.8
35	5.1	6.4	7.6	10.2	12.7	15.3	17.9	20.4
36	5.2	6.5	7.8	10.5	13.1	15.7	18.3	20.9
37	5.4	6.7	8.1	10.8	13.5	16.2	18.9	21.5
38	5.5	6.9	8.3	11.	13.8	16.6	19.3	22.1
39	5.7	7.1	8.5	11.3	14.2	17.	19.9	22.7
40	5.8	7.3	8.7	11.6	14.6	17.5	20.4	23.3
42	6.1	7.6	9.2	12.2	15.3	18.2	21.4	24.3
44	6.4	8.	9.6	12.8	16.	19.2	22.4	25.6
46	6.7	8.4	10.	13.4	16.5	20.1	23.4	26.8
48	7.	8.8	10.4	14.	17.4	21.	24.4	28.
50	7.2	9.	10.9	14.6	18.2	21.8	25.4	29.
54	7.8	9.8	11.8	15.6	19.6	23.6	26.4	31.2
60	8.8	10.8	13.1	17.4	21.8	26.2	30.6	34.8

Diameter Pulley.	Width of Belt in Inches.							
	18	20	22	24	26	28	30	32
In.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.
24	16	17	19	21	23	24	26	28
30	19	22	24	26	28	31	33	35
36	24	26	29	31	34	37	39	42
38	25	28	30	33	36	39	41	44
40	26	29	32	35	38	41	44	47
42	28	31	34	36	40	43	46	49
44	29	32	35	38	42	45	48	51
48	31	35	38	42	45	49	52	56
50	33	36	40	44	47	51	54	58
54	35	39	43	47	50	53	58	62
60	39	44	48	52	57	61	65	70
66	43	48	53	58	62	67	72	77
72	47	52	58	63	68	73	78	84
78	51	57	62	68	74	80	85	91
84	55	61	67	73	79	86	91	97
96	63	70	76	84	90	98	104	112
120	78	88	96	104	114	122	130	140
144	94	104	116	126	136	146	156	166

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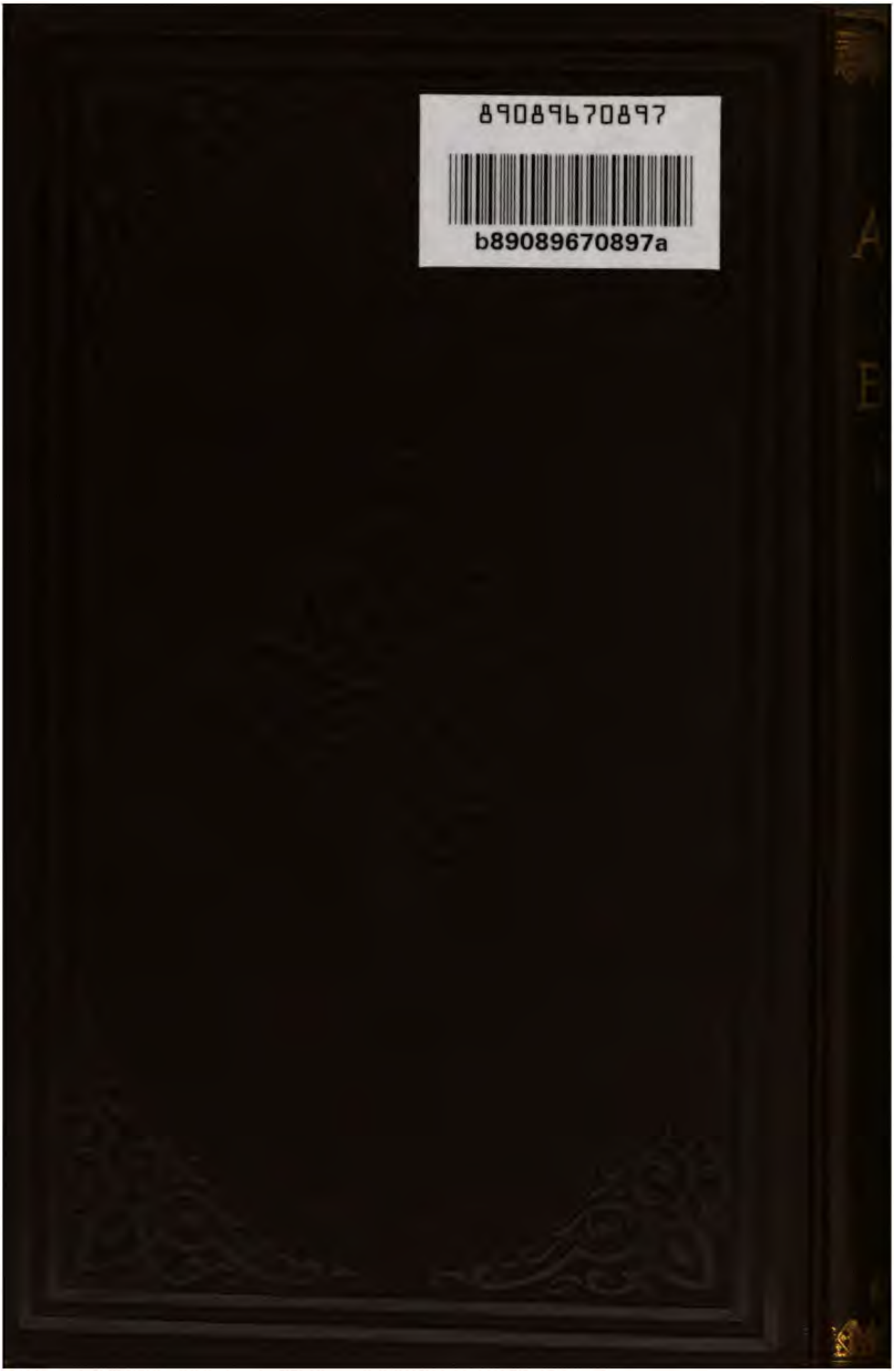

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